NEERING
IBRARY

MACHINE DESIGN

May 1946

In This Issue:

Front or Rear Engine Autos?

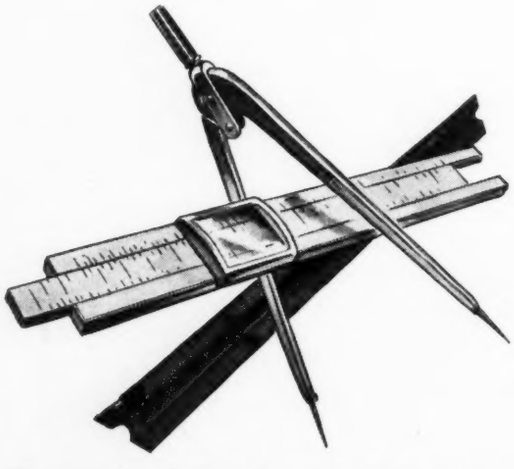
Combating Bearing Failures

Ever Figure What a 765 hp Motor Throws Off in Heat?

Our Navy Did... on fleet tugs down in the tropics, where engine room temperatures were running high as 140°F! In 25 by 30 ft of space, that's like having 140, thousand-watt radiant heaters "raising blisters" on you all at one time!

Here's How Navy Figured: To get 765 shaft hp out of 94% efficient propulsion motors, of which there are four aboard, they were putting in about 812 hp of electrical energy. Or, 47 hp went in that didn't come out of the shaft, but escaped through the frame. This ran about 35 kw heat per motor.

Well, Our Temperatures Began To Rise when Navy laid down its 'specs'. "Design a marine water-cooled motor—make it fit into the floor space of your open drip-proof type—no major changes in ship design. Make it extra-dependable, as fireproof as practicable".



We Measured, and figured, and made designs—used a new, toxic gas-free Glass Melamine insulation that required extensive changes in design and fabrication. We put special air passages inside the motor—yet kept accessibility. We saved 25% space over our previous commercial marine design by figuring a special shape cooler—without cutting down cooler efficiency.

Then We Made Those Motors Shock-Proof — by cutting out all cast iron, re-designing structural members for greater strength, resorting to modern, proven fabrication throughout. Result: A new kind of water-cooled motor that put a "tough" Navy job on ice, and also proved a point! A 1999

There's a Point to this story: Every time Allis-Chalmers engineering discovers new ways of solving *special* motor problems, like this one, it also learns how to build better *standard* motors for you! Watch for these new and better motors from A-C. ALLIS-CHALMERS, MILWAUKEE 1, WIS.

Wait 'til you see the NEW

ALLIS-CHALMERS MOTORS!

MACHINE DESIGN

THE PROFESSIONAL JOURNAL OF CHIEF ENGINEERS AND DESIGNERS

EDITORIAL STAFF

Laurence E. Jermy, Editor

Associate Editors

John W Greve — Colin Carmichael

Richard K. Lotz — Roger W. Bolz

H. N. Goga, Assistant Editor

Frank H. Burgess, Art Editor

New York.....B. K. Price, L. E. Browne
Chicago.....Erle F. Ross
Pittsburgh.....J. C. Sullivan
Detroit.....A. H. Allen
Washington.....L. M. Lamm
London.....Vincent Delport

BUSINESS STAFF

G. O. Hays, Business Manager

Mary L. Callahan, Advertising Service

Chicago.....H. H. Dreyer, Manager
New York.....R. H. Smith, Manager
E. S. Lawson, Asst. Manager
Cleveland.....H. B. Veith, Manager
Los Angeles.....F. J. Fuller, Manager

Research and Promotion

I. W. Zuber, Manager

Laura P. Stead, Circulation

MAIN OFFICE

Penton Building, Cleveland 13, Ohio

BRANCH OFFICES

New York 17.....16 East 43rd St.
Chicago 11.....520 North Michigan Ave.
Pittsburgh 19....2800 Koppers Building
Detroit 2.....6560 Cass Ave.
Washington 4...956 National Press Bldg.
Los Angeles 4 130 N. New Hampshire Ave.
London, S. W. 1..2 Caxton St., Westminster

Published by The Penton Publishing Co. E. L. Shaner, Pres. and Treas.; G. O. Hays, Vice Pres. and Gen. Mgr.; R. C. Jaenke, Vice Pres.; F. G. Steinebach, Vice Pres. and Secy.; E. L. Werner, Asst. Treas. Published on seventh of month.

Subscription in U.S. and possessions, Canada, Cuba, Mexico, Central and South America: Two years, $10; one year, $6. Single copies, 50cents. Other countries: Two years, $14; one year, $8.

Copyright 1946 by The Penton Publishing Co.

CONTENTS

May, 1946 — Volume 18, Number 5

Cover—Postal Meter (Couretsy Commercial Controls Co.)

Itemized Index 7

Topics 100

Comparing the Merits of Front and Rear Engine Autos—Part I 103
By Austin M. Wolf

Scanning the Field for Ideas 109

Bearing Failures—Causes and Cures—Part I 113
By R. W. Dayton and R. E. Adams

Production Processes—Part XI—Contour Sawing 117
By Roger W. Bolz

Simplifying the Construction of Nomographs 123
By E. B. Steinberg

Mechanical Torque-Limiting Devices 127
By N. C. Wyeth and A. F. Gagne

Localized Surfacing Combats Wear, Corrosion 131
By A. R. Lytle

An Aluminum Spring May Be the Answer! 137
By Amiel Gelb

Trends in Automotive Transmissions—Part II 141
By Colin Carmichael

Lost Wax Investment Castings 146

Evolution of a Modern Transport Airplane 148
By Maxwell B. Bassett

Automatic Tension Control Improves Production 153
By Joseph Gepfert

Customer Acceptance Needs Emphasis (Editorial) 155

Outstanding Designs 156

Applications of Engineering Parts, Materials and Processes 160

Lubricant Selection Simplified (Data Sheet) 161

New Parts, Materials and Equipment 166

Men of Machines 176

Noteworthy Patents 182

Business and Sales Briefs 184

New Machines 192

Calendar of Meetings 194

Helpful Literature 291

OF all machine manufacturers:

66.3% will use Aluminum Alloys in their products in 1946, and **66.8%** of those who have used aluminum will use *more per machine* than prewar.*

90.7% will use sand castings, and of these **22.8%** will use *more per machine* than prewar. **48.1%** will use die castings, and of these **59.4%** will use *more per machine* than prewar.*

These data chart a trend which cannot be overlooked. They spell progress—modernization.

Have *you* any question regarding Aluminum Casting Alloys for *your* designs?

The members of this group invite inquiries regarding these alloys and their application—perhaps we can help you.

**From a survey of design engineering executives in 1561 plants, made by Machine Design Magazine.*

ALUMINUM RESEARCH INSTITUTE

111 West Washington Street, Chicago 2, Illinois

MEMBER ALUMINUM RESEARCH INSTITUTE

Samuel Greenfield Co., Inc.
Buffalo 12, New York

William F. Jobbins, Inc.
Aurora, Illinois

R. Lavin & Sons, Inc.
Chicago 23, Illinois

The National Smelting Co.
Cleveland 5, Ohio

Niagara Falls Smelting & Refining Corp.
Buffalo 17, New York

North American Smelting Co.
Tioga and Edgmont Sts. Philadelphia, 34, Pa.

Sonken-Galamba Corporation
Kansas City 18, Kansas

U. S. Reduction Co.
East Chicago, Indiana

Aluminum and Magnesium, Inc.
Sandusky, Ohio

The American Metal Co., Ltd.
New York City 6

Apex Smelting Co.
Chicago 12, Illinois

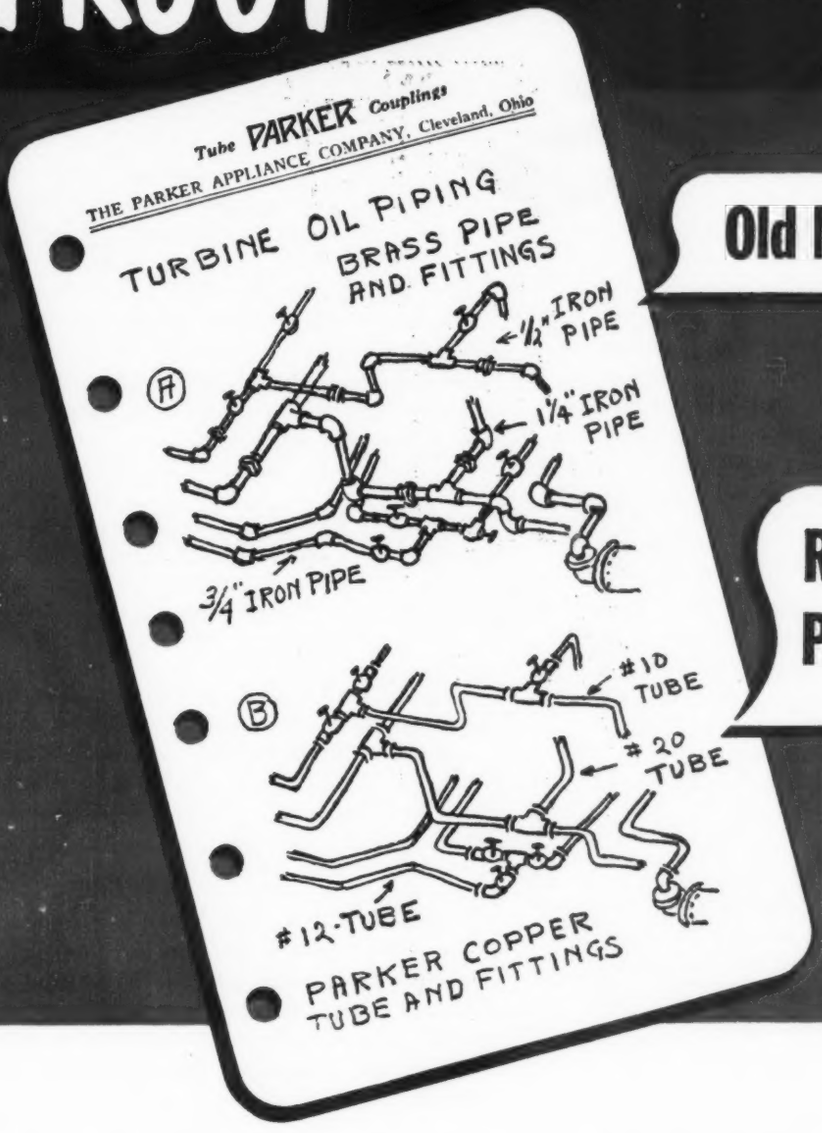
Berg Metals Corporation
Los Angeles 11, California

The Cleveland Electro Metals Co.
Cleveland 13, Ohio

Federated Metals Division American Smelting & Refining Company
New York City 5 and Branches

General Smelting Company
Philadelphia 34, Pennsylvania

PROOF FROM A PARKER ENGINEER'S NOTEBOOK



You can save time, money and materials with Parker Fluid Power Engineering.

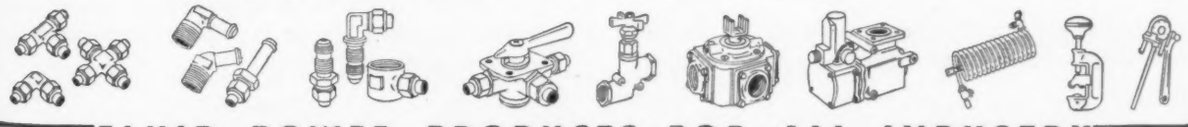
The actual case record pictured is typical. Our recommendation enabled this manufacturer to cut down on number of connections, on space and avoid leakage. Moreover, his customer got a better job—one that operates more efficiently, is easier to service.

We've solved many a manufacturer's problems with equally simple recommendations—using standard Parker equipment. For unusual conditions of design, where stock parts will not serve, we are able to supply "custom-bilt" valves and couplings. In every case the result has been the same—a simplified installation. May we show you how Parker Fluid Power Engineering can do the same for you?

Stocks Available at your Distributor's and at Parker Warehouses

THE PARKER APPLIANCE CO.

CLEVELAND • LOS ANGELES



FLUID POWER PRODUCTS FOR ALL INDUSTRY

Topics

NEW TYPE of coil spring composed of five wires, one serving as a central core with the others wound about it, has potential value for engine valves. Developed in a German technical institute, the spring uses 20 per cent less material than a solid-wire spring of the same rate and has a 20 per cent greater fatigue life.

GRAPHITE ANODES for electronic tubes have been perfected by United Electronics Co. whose research has eliminated the former limitations of graphite for internal anodes. Impregnated with zirconium and utilized in conjunction with an "isolated getter trap", the anodes do not deposit on the glass tubes, a characteristic which has been a barrier to their heat dissipating capacity.

SECOND ONLY to the loss of human life by our American forces in World War II was the tragic cost in our mineral wealth. According to the *Army Ordnance*, the military mind as well as the industrial mind must be taught that waste in peace or in war is a calamity of the first order.

BY REDUCING FUEL consumption from 20 to 50 per cent, the high pressure steam turbine developed by U. S. Steel's Federal shipyard increased the distance a ship can travel without refueling—a factor of incalculable importance while fighting Japan across the vast reaches of the Pacific.

REMOTELY CONTROLLED, a Lockheed P-80 will be given a series of power-dive tests at Muroc Army Air Field, Calif., to prove the maximum speed at which the plane can be flown safely. In this way danger to the pilot is eliminated and the plane can be put through paces never before attempted.

IMPURITIES in the basic materials of plastics, believed to cause present breakage, wear and deficient insulating properties, should be largely eliminated through the use of the pearl polymerization method, according to Dr. W. P. Hohenstein, of the Polytechnic Institute of Brooklyn. The process produces a basic plastic and consists of dispersing a monomer in a nonsolvent medium by mechanical agitation, converting the monomer globules to the polymer under the influence of heat and catalysts.

ONE-FIFTH SECOND is the life of a motor designed to bring a gyroscope to speed in an electric torpedo. The motor has an instantaneous rating of 22 horsepower and weighs 8½ pounds.

POTENTIALLY THE WORLD'S most universally available metal is magnesium yet all of it produced in the world to date could have been taken from about one-tenth of a cubic mile of seawater. Significant is the fact that no limited resources are being depleted by the use of magnesium.

SYNTHETIC MICA, perfected in the laboratories of the KWI Ceramics Institute in Germany, is reported by U. S. investigators to be as good as natural mica. The synthetic consists of mixed oxides, fluorides and silico-fluorides containing such metals as aluminum, magnesium, iron, chromium, and vanadium.

IF ATOMIC ENERGY were obtainable absolutely free today and if every pound of coal burned to make electricity were replaced by atomic fuels, the resulting reduction in the delivered cost of electricity could not exceed 15 per cent. Total saving would be less than ½ per cent of the national income.

SELENIUM RECTIFIERS now may be submerged in boiling water or packed in ice and yet convert alternating current into direct current at an efficiency equal to that obtained under normal operating conditions. Developed by Westinghouse, the rectifier consists of a tin-plated, hermetically-sealed container inside of which is selenium suspended in oil.

MACHINE DESIGN

Comparing the Merits of Front and Rear engine autos

By Austin M. Wolf
Automotive Consultant
New York City

Part I

CONVENTIONAL automobiles of today, insofar as the arrangement of their power plants, transmissions and drive wheels are concerned, are imitations of a European design pioneered by Mercedes. It was the improved accessibility of the engine which caused early American manufacturers to adopt this arrangement and to abandon their underbody location of power plant, which invariably used a chain drive to the rear axle from a planetary gear set in line with a transverse crankshaft.

FRONT DRIVE HISTORY: Application of power to the front wheels of an automobile is by no means new. In 1898 electric (storage battery) hansom cabs were used on New York City streets with an electric motor driving each fixed front wheel, with steering action obtained by deflecting the rear wheels. Woe unto the driver if he attempted to drive ahead and away from a high curb! Walter Christie piloted a front-drive racing car over the Daytona Beach sands and on dirt racing tracks. He also built an experimental front-drive taxicab. In late years, the Ruxton and Cord front-drives appeared in this country, as well as Citroen in France and Alvis in England and a few front-drive racing cars over here. The latest exponent of the front-drive principle is the recently announced Kaiser car.

REAR DRIVE: Shortly after World War I, rear-engined rear-drive cars began to appear. One of

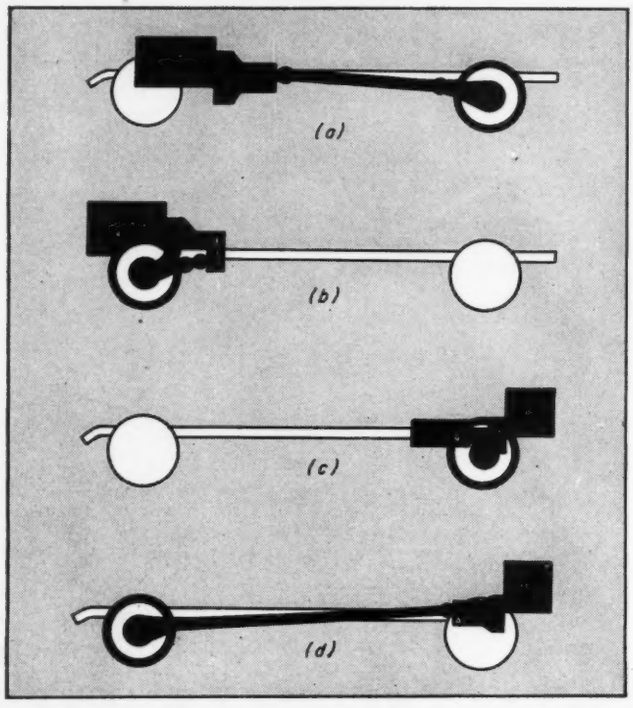


Fig. 1—Auto drive-component arrangements are: (a) Conventional, (b) front-engined, front-drive, (c) rear-engined, rear-drive, and (d) rear-engined, front-drive

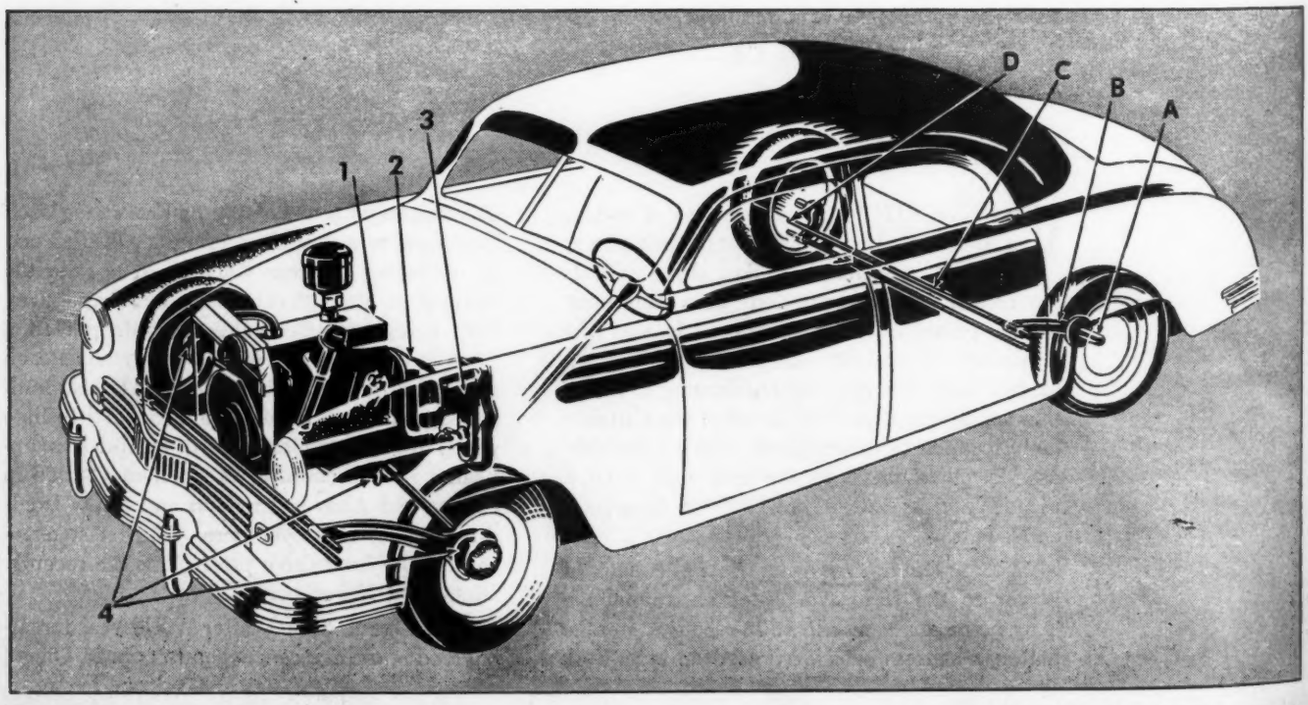
the earliest was that of Rumpler in Germany who also introduced the "swinging" driving axle. In the 1920's, the Tatra of Czecho-Slovakia appeared, Mercedes built a small car and the American bus builders "went at it" in earnest in the 1930's, including GMC, Twin Coach and Mack. Bill Stout built his Scarab in this same period as a modern rear-engined rear-driven car, though it will be recalled that around 1908 the Adams-Farwell appeared with a single cross-seat, open body in whose rear deck was located a revolving-cylinder air-cooled radial engine (forerunner of the French Gnome aviation engine of World War I) with its stationary crankshaft in a vertical position.

VEHICLE CLASSIFICATIONS: In addition to the three classifications thus far mentioned, i.e., (*a*) The conventional type of construction, (*b*) front-engined, front-drive and (*c*) the rear-engined, rear-drive, there is (*d*) a front-driven, rear-engined job. This is the conventional car with its essential components reversed and with the driving axle provided with steering heads, or the latter replaced by transverse universally-jointed shafts. Only a few experimental vehicles of this type have been built, yet quite a few appear in the patent records. There are not sufficient virtues to this type to discuss it further. Referring to *Fig.* 1, these four classifications are shown diagrammatically as (*a*), (*b*), (*c*) and (*d*) respectively, with the power plant and driving mechanism shaded black.

Early automobiles were built very high, chiefly to obtain road clearance on rutty roads. Present concrete roads have permitted reduction of clearance height, to the dismay, however, of many a driver who has had to leave the improved road and follow a country trail and got "hung up". The trend has been to keep lowering the center of gravity for improved riding qualities and increased safety; so much so, that one has to be an acrobat and a contortionist to enter a modern sedan from a high curb. However, a low design is classy in looks, (how well do the body designers and sales departments know it) and we are all prone to sacrifice something in the way of comfort for that quality with cars as well as with other items. Naturally a vehicle looks much longer as the height-length ratio becomes less.

FLOORS AND TUNNELS: In the old days, while the typical owner knew that there was mechanism below the flat floor, he was not conscious of anything being there. Today we have a "tunnel" in the floor of the rear compartment in

Fig. 2—In Kaiser car the engine 1, clutch 2, and transmission 3, are up front along with propeller shafts 4 which drive the front wheels. Fixed at D, torsion bar C for left wheel suspension mounts trailing arm B, at opposite end of which is spindle A which carries the wheel



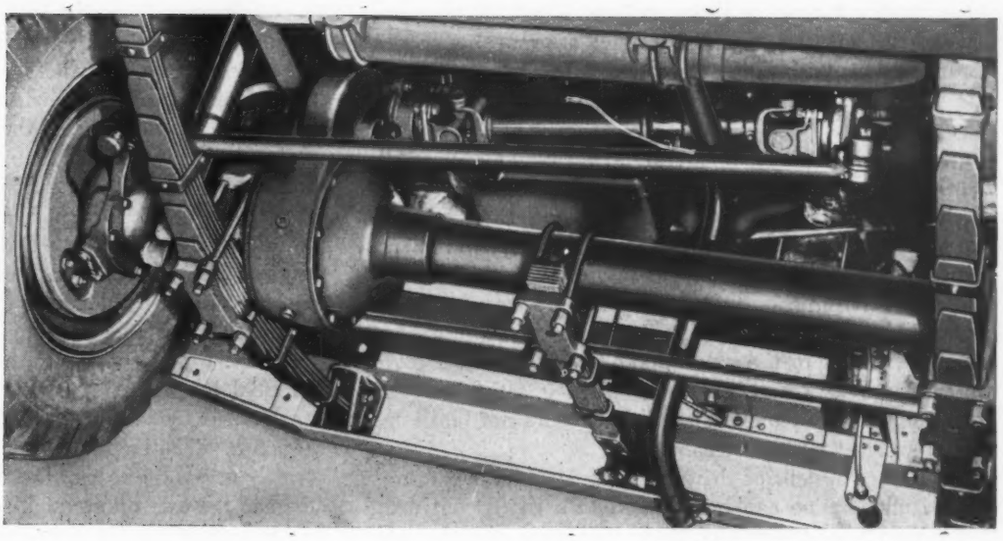


Fig. 3—View looking up into front-drive components of the new Marmon - Herrington Delivr-All truck. Clearly seen are the rigid driving axle, the short universal - jointed propeller shaft and differential, and (left) one of the two steering heads

order to get the necessary low floor level and at the same time low seat height, to be able to place a low roof line over the passenger's head at a minimum distance above the ground. The automobile manufacturers have never declared their thanks openly to the passing of the silk high hat vogue. The automobile surely forced its obsolescence.

Hypoid rear axle gearing helped in dropping the propeller shaft to a lower position. Where there are no "tunnels" or only modest ones, the floor board or steel sheet slopes from the center slightly downward toward the body sides to make the necessary propeller shaft clearance, required with the up-and-down movement of the rear axle, less obvious. Of course, if this longitudinal propeller shaft could be discarded, the floor could be dropped as low as might be desired, except as dictated by ground clearance. Herein lies one of the main virtues of our second classification for the vehicle components' arrangement for the front-engined, front-drive as well as for the third classification for the rear-engined rear-drive. The body designer is practically unfettered. This could also be obtained in a car having full electric transmission with the engine driving a direct-coupled generator and with a driving motor to the rear of the rear axle or individual motors at or close to each rear wheel. However, the high cost of such an arrangement, excessive weight and its relatively low efficiency for the usual passenger car precludes it for consideration. The same would hold true for a full hydraulic system.

The Case for Compactness

COMPACTNESS OF COMPONENTS' ASSEMBLY: Proponents of classifications *b* and *c* state that a compact unit is always more efficient than one in which the components are "spread out all over the lot", the latter referring, of course, to the customary construction. No one could argue on this point in the case of steam power generating plants, for instance. In motor vehicles this can be true, yet it is not always so. The long propeller shaft of the regular car is the component with the biggest "spread" and it must be built with a large diameter tube between the universal joints to prevent "whipping" at high speeds when the rear axle bounces over road obstructions. Such whip, if it does occur, absorbs power in creating vibration. The bearing loads in the universal joints are greater with a heavier shaft, yet with low angular joint deflection as is the case with a long shaft, the power loss is extremely small. In many designs of the *b* and *c* classifications, short transverse universally jointed shafts are used between the frame-mounted differential and the wheel. Here, in spite of low weight, joint angularity could be far more serious than the previously mentioned joint-bearing loading.

FRONT DRIVE EXAMPLES: It will be well to examine a few actual constructions before entering into a detailed analysis of the relative merits of the two general classifications under consideration. A phantom perspective view of the Kaiser car is shown in *Fig.* 2. In this, front drive and front location of the power plant 1, clutch 2 and transmission 3 are combined. The differential is located below the rear end of the crankcase and just ahead of the bell-housing. The sump stamping at this point is as shallow as possible to permit nesting the differential case as high as possible. Engine, clutch and transmission are conventional units, assembled in the usual manner. A drop gear is installed back of the transmission with a close-coupled double universal joint between it and the differential case which is supported from the bottom of the bell-housing. This calls for a raised engine location; hence the cooling fan is mounted on the front end of the crankshaft. The engine is foremost in order to place more weight on the front wheels than in previous designs.

It will be recalled that the Cord car had the differential between the clutch and the transmission but with the engine to the rear, thus bringing the transmission ahead of the "axle". The Kaiser has the front wheels mounted on pivoted trailing arms which was also the case with the Cord, but the former incorporates torsion bar suspension. Front bars correspond with rear bars in which *C*, the upper, is fixed at *D* and twists as the left wheel load varies. The wheel is mounted on spindle *A* in the end of the trailing arm *B*, whose pivotal center receives the left end of the torsion bar. The front trailing arm terminates in a steering head.

Drive from the differential to each wheel is through a universally jointed propeller shaft 4 at each side. There is no axle in the usual sense, nor is there any axle with individual wheel suspension in most of the present, conventional cars. In European terminology, "axle" embraces

either the rigid type or those systems in which the suspension system, composed of trailing arms or transverse springs and/or wishbones, takes the place of the previous axle bed for a front axle or the axle housing for a driving axle.

The newly introduced Marmon-Herrington Delivr-All door-to-door delivery truck, shown in *Fig.* 3 (looking upwardly from vehicle front) utilizes a rigid front driving axle with steering heads and the transverse power plant located above it and at the right side of the vehicle, with clutch and transmission to the left. A drop case from the transmission encloses a gear train and a short propeller shaft connects it to the axle differential drive. The high center of gravity of such a power plant is satisfactory in the case of a slowly moving truck but would not be desirable in a high-speed passenger car.

The Kaiser system of driving forward under the transmission and engine can be adapted to a rigid front axle by raising the engine to give riding clearance over the differential bowl (if they are centrally aligned; otherwise the necessary axle clearance must be provided at the points of nearest approach) and lengthening the propeller shaft to cut down joint angularity. In fact this would practically be the Jeep design with the rear drive omitted. The entire mechanism can also be swung around to the back of the vehicle and coupled to a regular rear axle.

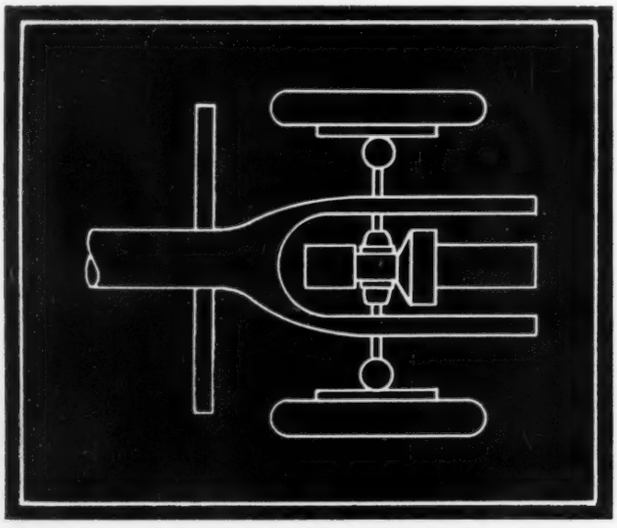


Fig. 4—Arrangement of drive components in a typical rear-driven, rear-engined automobile

REAR-DRIVEN REAR-ENGINED VEHICLES: There are no rear-driven rear-engined cars built in America but a number have been built in Europe such as Tatra, Volkswagen, Hansa Lloyd, etc., in which a tubular backbone frame terminates in a fork encompassing the rear power plant and transmission units as shown diagrammatically in *Fig.* 4. Transverse propeller shafts are used, giving in reality a duplication of *Fig.* 1 *b* with the chassis swung around but with no steering action of the wheels. Mercedes utilizes the same arrangement, except that the open double-jointed propeller shafts and separately supported wheels are replaced by swinging-axle "halves". These are anchored at their inner ends concentrically with the universal joint center adjacent to the differential and the wheel is mounted on the outer extremity of each "half".

Our country leads the world in the production of rear-engined busses. In the early 1930's, General Motors Coach produced a vehicle with the engine, clutch and transmission as a unit which was supported in a subframe removable from the rear, the engine being rearmost. A pinion and ring gear with the differential occupied a case bolted to the front of the transmission. A transverse, universally jointed drive shaft at each side actuated a rear wheel, thus constituting a DeDion drive (termed after the original DeDion Bouton voiturette which incorporated this design) since a dead axle of tubular form had spindle ends on which the wheels were mounted. The axle carried the load and the transverse shafts to the frame-mounted differential transmitted the power to the wheels. This was superseded by the angle-drive shown diagrammatically in *Fig.* 5 *a*. The engine, clutch and transmission run transversely across the rear of the bus, with a diagonal propeller shaft to the axle differential.

Mack developed a right-angle drive, *Fig.* 5 *b*. A bevel gear on the clutch shaft actuated the upper transmission shaft to its rear which in turn drove a lower shaft through the selected gears and the power was then transmitted forward to the rear axle through a propeller shaft. Twin Coach used the system shown in *Fig.* 5 *c* in which the transmission case was bolted directly to the transverse engine, the clutch being located to the left of the transmission. Power returned to the right through a quill shaft surrounding the clutch shaft, on which transmission gears were mounted; they drove down to a countershaft which terminated at its right end in a bevel gear, whose mating gear drove forward through the propeller shaft. Twin Coach later raised the engine and brought the rear axle center to almost below crankshaft center and drove the vertical axle pinion shaft through an upright telescoping shaft to take care of the up-and-down axle movement, including, of course, a universal joint at each end. At the same time, the clutch was moved back to the conventional flywheel location. In all the examples of *Fig.* 5, the differential is offset to one side of center.

Beck bus uses standard units back of the driving axle including, proceeding rearward, a short propeller shaft, transmission, clutch, engine and radiator. In the Ford Transit Bus the common axis of the engine, clutch and transmission runs longitudinally across the rear with a 90-degree bevel gear set in back of the transmission (in reality at the right side thereof) with the drive forward through a short propeller shaft to the right of the longitudinal vehicle center. In this way the convex clearance plate in the floor over the differential bowl is offset close to a wheel-house seat where it will not be tripped on, which would be the case if it were in the center.

Factors Affecting Placement of Weight

MASS DISTRIBUTION: Since the automobile is an instrument of mobility, distribution of its weight is a most important factor. With the greatly increased facilities for producing aluminum, there are high hopes in some quarters that weights can be reduced considerably, particularly in engines. An aluminum engine, other than cylinder liners, piston rings, connecting rods, crankshaft and flywheel, camshaft, valves and a few other incidentals, is not a novelty since the last Marmon cars were so equipped. In the early 1920's considerable experimental work was

done by the Aluminum Company of America. It will be readily seen that were a power plant to weigh practically nothing, there would be no need to worry whether it were placed on the roof or to the extreme front or extreme rear. Would that that were true but, unfortunately, the engine represents considerable mass.

To get sufficient weight on the front driving wheels, practically all recent designs here and abroad have the power plant or a major portion thereof ahead of the suspension system and/or "axle". With a vehicle on an upgrade, as indicated in *Fig.* 6, weight is transferred from the front to the rear wheels as compared with level ground distribution and in the case of the front drive, this is not desirable. To prevent loss of traction under such conditions, power plants are placed well forward so that front wheel loading will be sufficient in climbing hills. With the center of gravity thus located farther forward, however, the car does not fare so well in braking. Greater weight is transferred to the front wheels and the present nosing-down tendency of the front end of the present vehicle would be further accentuated.

In the conventional car or with the rear-driven rear-engined car, the weight automatically increases on the driving wheels on a grade. It will be readily seen that there is a limit to how far back the power plant and transmission machinery can be placed in the latter since on a steep grade there might not be sufficient weight on the front wheels for steering response. Similarly, under snow or icy conditions when on the level, lack of steering might be encountered.

Another factor that must be considered is the ability of the vehicle to retain a straight course and avoid sluing. With the center of gravity further to the rear as in the case of *Fig.* 1 *c*, there would be a greater tendency toward skidding under slippery road conditions with the center further back in relation to the points of support (the wheels on the road) since the c.g., under unstable conditions, would try to swing around ahead of the support, whereas the forward center of gravity location symbolized by *Fig.* 1 *b* would lend stability. Thus far this factor has been considered by itself but, simultaneously, nosing-down proclivities must be given consideration. Under severe braking, a center of gravity well forward would result in relieving weight from the rear wheels, lessening their braking effectiveness due to forward weight transfer as well as the nosing-down tendency. Weight transfer when braking is recognized in present day cars with 60 per cent of the braking effort being exerted on the front wheels and 40 per cent on the rear. Greater front weight calls for increased front braking.

Fig. 5—Rear-driven, rear-engined bus arrangements, all having transversely mounted engines. (a) angle-drive, (b) transmission at right angle to clutch and (c) clutch at extreme left of engine-transmission-clutch unit

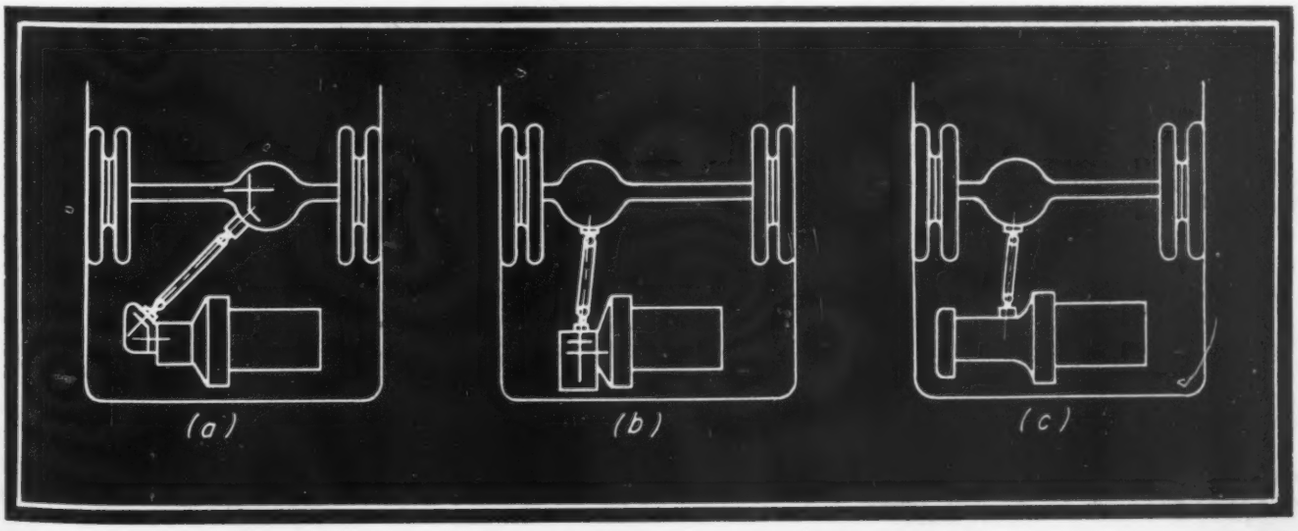
Front Drive An Asset on Fast Turns

In view of the sluing proclivities mentioned in the last paragraph, the opposite condition for high maintained speed resulted in the building of a few racing cars with front drive. "Gunning the car" on the straight-away, into a turn or around it, gave the driver a sense of security and positive directional stability. Here the front driving wheels were "dragging" the center of gravity behind them, making for a stable situation while the power was "on". Unfortunately we are unable to utilize this method on the highways since we cannot tramp on the accelerator when we are seeking safety and security.

From the foregoing discussion, it is obvious that mass distribution and its weight transference and inertia effects under the many varied influences, most of which have been mentioned, afford a highly complex problem, varying with type of design and also dimensional disposition in all planes. Each particular design is a case unto itself.

Cooling: The front-engined cars, whether of classifications *a* or *b*, have the radiator at the front of the vehicle in the stereotyped location so that it would be swept by the fan draft and the flow induced by the vehicle's motion. Should an air-cooled engine be used, the problem would again be simple in view of taking in air ahead of the vehicle.

With the water-cooled rear engine and a rear radiator location, it is not as simple a matter to convey the air through the radiator core. The body contour and particularly its form over the rear portion can be conducive to flowing air into the engine compartment at speed but the range of speed at which this condition would be efficient would be rather limited. The problem, however, should not be insuperable inasmuch as Tatra is utilizing a rear



air-cooled engine. It is, of course, possible with water cooling to place the radiator at the front of the vehicle but this results in a lot of "plumbing" as well as adding to the capacity of the system. Nevertheless, the rear engine location calls for considerable care and possibly expense in the cooling system layout.

HEAT, FUMES AND NOISE: During summer weather or in a hot climate, the conventional forwardly located engine pours out heat which can be quite uncomfortable despite the carefully insulated dash and toe boards. Considerable heat comes back along the sides of the car and under it. The sheet metal under the fenders usually is provided with louvers or ducts to exhaust the engine compartment in addition to what air goes out of the hood side and under the car. With the rear power plant, conditions would be better at most times. Considerable insulation must be placed at the back of the rear seat (or under it as well, as in the case of the rear-engined busses) to protect the occupants. Outside air must be induced into the engine compartment to prevent a stagnant space at the forward

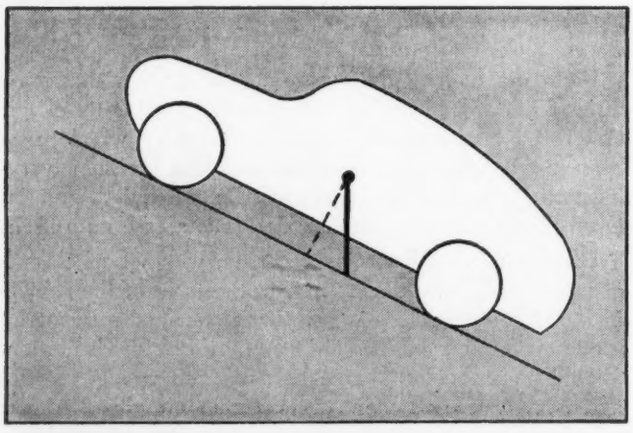


Fig. 6—On upgrade, weight shift is to rear wheels, giving them better traction than front wheels

end. With little or no vehicle speed there would be little choice, much depending upon the prevailing wind.

The matter of fumes would follow to a large extent the above discussion on heat. At speed, a vacuum can be easily induced in the body at a point of low air pressure and this calls for thorough isolation of the engine compartment regardless of its location. Extraneous heat and fumes might conceivably be more easily picked up with the front engine location. The designer of the ventilating system must bear this in mind with reference to fresh air inlets.

On the question of noise, the occupants nearest the engine would hear it more than the occupants of the other seat. With the engine at the rear, the driver and the passenger alongside of him would hear little noise but those on the back seat would hear considerably more since they are seated closer to the engine with their backs very near to it and particularly their ears, in view of their inclined body posture, instead of being spaced from the engine by the length of their legs. Strictly speaking, the operator should be close to the engine in order to hear any unnatural sounds which might indicate that something is wrong. Considerable trouble was had with some of the early rear-engined busses due to the fact that the driver did not hear the gears shift and he was not able to synchronize his clutch actuation therewith. Microphones were installed in the engine compartment with a speaker close to the driver's ear. However, the general public is not mechanically minded and in most instances it would probably mean nothing whether a noise were heard or not.

In his concluding article, Mr. Wolf will discuss other pertinent phases of the subject, such as torque reaction, driver's position, engine arrangement, controls and instruments, suspension and drive, etc.

Single Materials Specification System Would Have To Be Flexible

To the Editor:

I would like to contribute the following comments on the symposium "Should a Single Materials Specification System Be Established?" which appeared in the March issue of MACHINE DESIGN.

During the war it was necessary for most manufacturers working on war contracts or subcontracts to purchase raw materials to a great variety of governmental and private specifications. At first blush, it would seem that a single series of specifications would be highly desirable and that this would eliminate considerable confusion. If, however, such a single system were inflexible, I do not think it would prove to be desirable. On the other hand, if the system were flexible enough to take care of most constructional materials requirements adequately, permitting special specifications for special cases, then a single system might be advantageous.

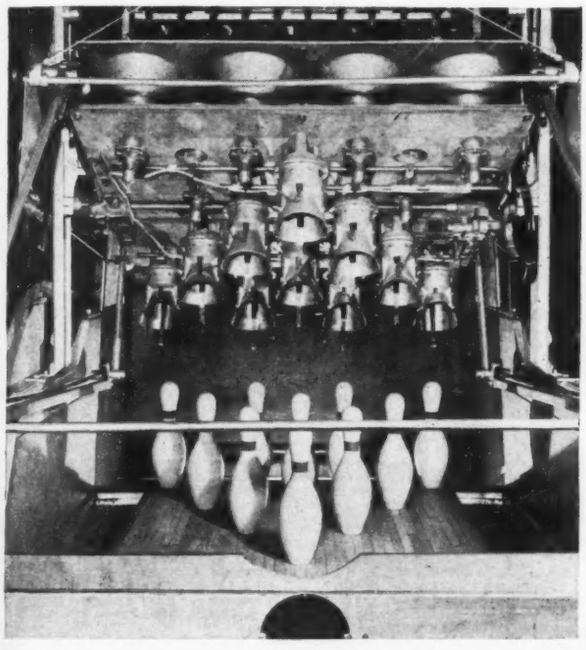
Evolution of a technical specification, at least in the work of the American Society for Testing Materials, follows the following general procedure: Methods of test are devised or developed, data are obtained on existing materials and then specification limits are agreed upon between producer and consumer. This results in a standard.

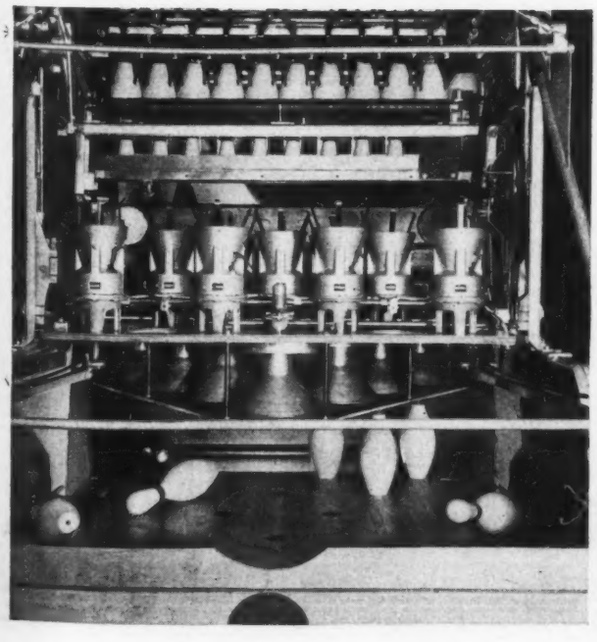
The purpose of the standard is to establish definite quality or performance levels. Actually, the user of materials may be interested at different times in a variety of quality levels. For example, the use of a laminated plastic in an airplane wing would require high quality material proved by rigorous tests, whereas the use of plastic in a tag would not require such high quality material. It seems desirable that technical standards be established for principal use construction materials. This is being done very well by ASTM and its specifications are sponsored to ASA for approval as national standards.

A variety of specifications come from governmental departments. The Army, Navy and Air Force feel that they need different qualities of material for the same purpose. It would be desirable for the various branches of the government to try to arrive at mutual standards. Some progress in this direction was made during the war and it is hoped these co-operative efforts will continue. However, the government is required to purchase materials from the cheapest bidder on a basis of *caveat emptor*. Industrial specifications, such as ASTM and ASA, are standards for the industry and mutually agreed upon by producer and consumer. There would be an advantage to the government to purchase to such specifications as far as its needs and the general situation will permit.

—J. R. TOWNSEND, *President*
American Society for Testing Materials

Scanning THE FIELD for Ideas





AUTOMATIC pin-spotter for bowling alleys, which is shown in the accompanying illustrations, performs all the intricate mechanical operations associated with returning the ball, sweeping "deadwood" off the alley, resetting the pins, and distinguishing between strikes and spares.

It is powered by two motors of less than two horsepower. One motor drives the mechanisms while the other is coupled to a pump supplying vacuum to suction cups for holding the pins in the machine. Relays governed by positioning of cam units as well as the positions of the ball and pins initiate the sequence of functions performed. Twenty-one pins are used in operating this machine which has been developed and perfected by American Machine & Foundry Co. After a set of pins has been spotted on the alley, the reversible turntable which placed them moves into place as shown in the view, left above.

In the second view, left below, the ball seen in the left background has brought the machine into operation by switch action caused by the weight of the ball in the pit. An elevator arrangement is starting to lift the ball on its return to the bowler while at the same time a guard has lowered in the foreground to ward off any vagrant ball. Also, the machine has lowered to pick up the standing pins by a suction cup arrangement. The standing pins are lifted while a "sweep" clears the alley of deadwood. The lifted pins are then returned to the exact position even though off spot, and the guard and sweep are lifted in readiness for the second ball. If the first ball is a strike, the machine senses this condition because no pins are standing to engage the suction cups. After the second ball or strike, the machine lowers to spot exactly ten pins for the next frame.

Fallen pins which have been swept into the pit are carried to the top of the machine by a conveyor and distributed by mechanical action in ten separate cups for subsequent spotting. For picking up pins that have remained standing after a ball has been delivered, ten

funnel-shaped guides on the reversible table steer the suction cups over the tops of the standing pins, grasp them and lift them out of the way. These grippers replace an off-spot pin exactly in the same position occupied before, as required by rules.

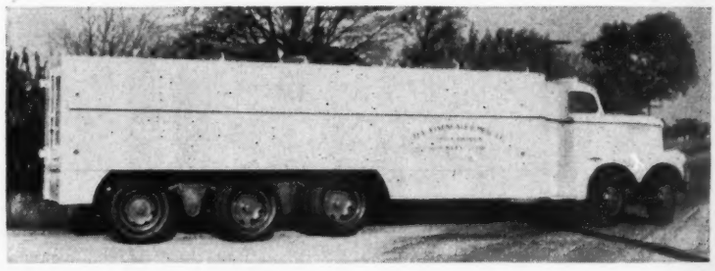
Four front wheels, in tandem relation rather than dual, provide easier steering for large trucks, obviating power steering, and allow more load to be carried on the front wheels. The added weight on these wheels in the truck illustrated at right results in better traction and reduced road shock as well as permitting more pay load on the truck.

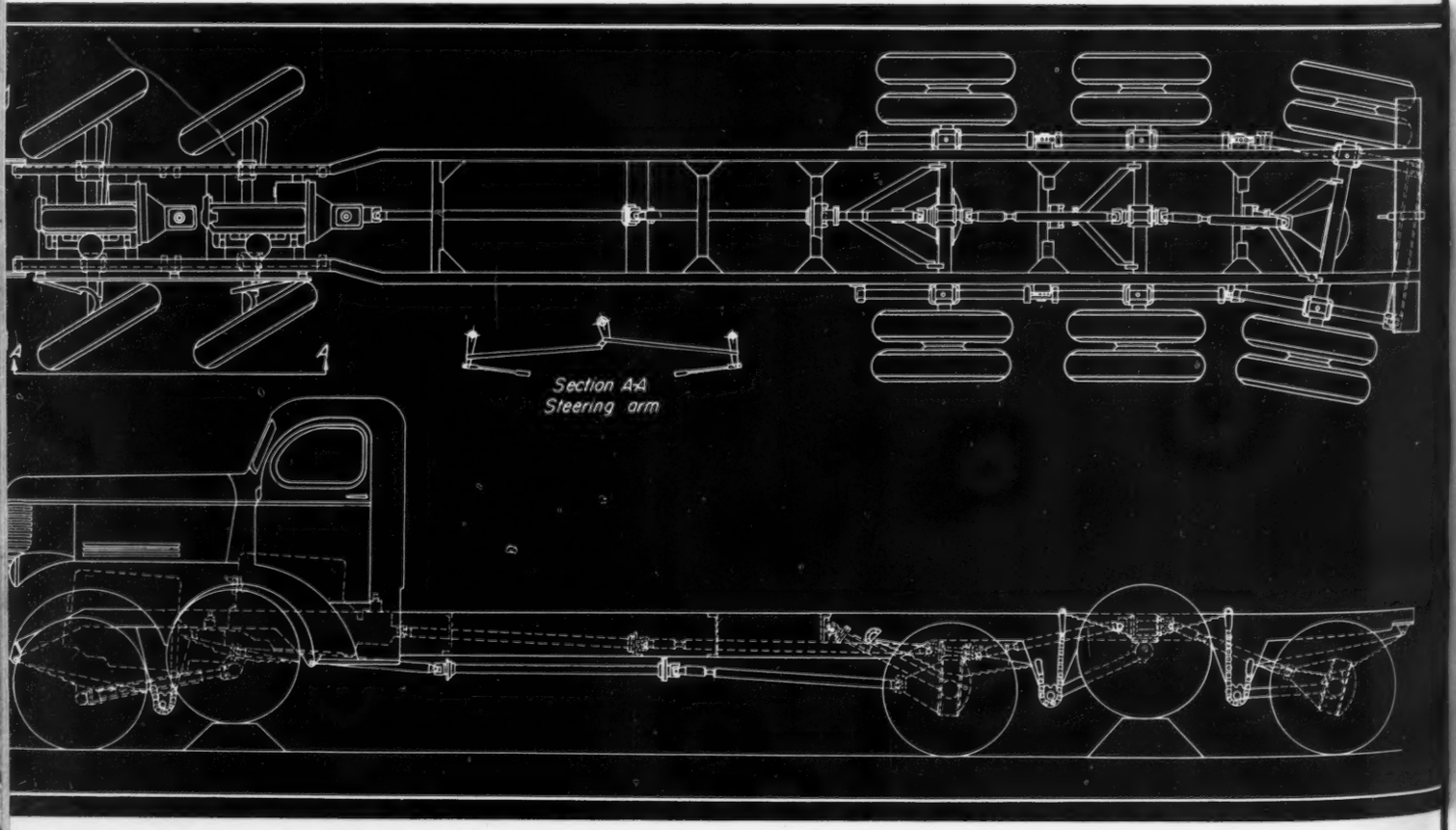
In the diagram below is shown the tandem steering arrangement and the steering linkage in section A-A. This linkage is conventional except that an additional link is utilized for the second set of wheels and is so proportioned as to allow for the difference in turning radii between the two axles.

The truck has three rear axles including two driving axles with a dead axle between. To help the front steering wheels turn on curves, the rear axle and wheels are pivoted to follow, thus preventing side skidding of the rear wheels. Springs of adjacent axles have compensating means which insure proper contact of the wheels under all road conditions.

Power for the truck is supplied from two standard engines mounted in line; one under the hood is direct-connected to the leading rear axle and the other beneath the cab drives the rear-most axle. The engines may be used together or independently. Power selection is accomplished by means of a differential type synchronizer and, when both engines are used, torque output is balanced between the two by a torque equalizer actuated by manifold pressure. Standard truck parts are used throughout with the exception of the frame and suspension to facilitate servicing and repair when necessary.

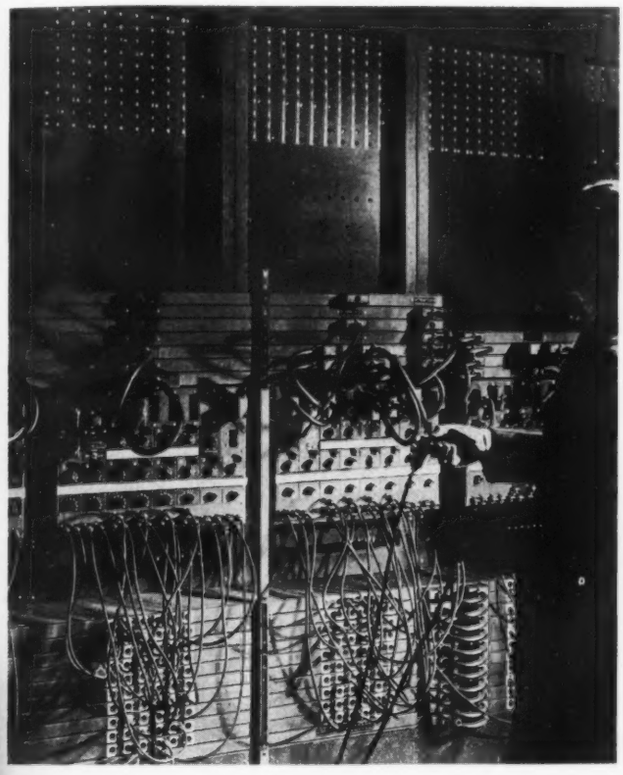
Calculation bottlenecks threatened by the cumbersome methods in computing firing tables for ordnance equipment led to the development of ENIAC (electronic numerical integrator and computer). Developed by the University of Pennsylvania for the Ordnance Department, this machine utilizes 18,000 electronic tubes and occupies a room 30 by





50 feet in size. Included in the circuits for the calculator, shown at bottom of page, are 500,000 soldered joints, 70,000 resistors and 10,000 capacitors. Accumulator panels used for adding or subtracting are shown below. The plug-in cables connect the panels together for different mathematical processes.

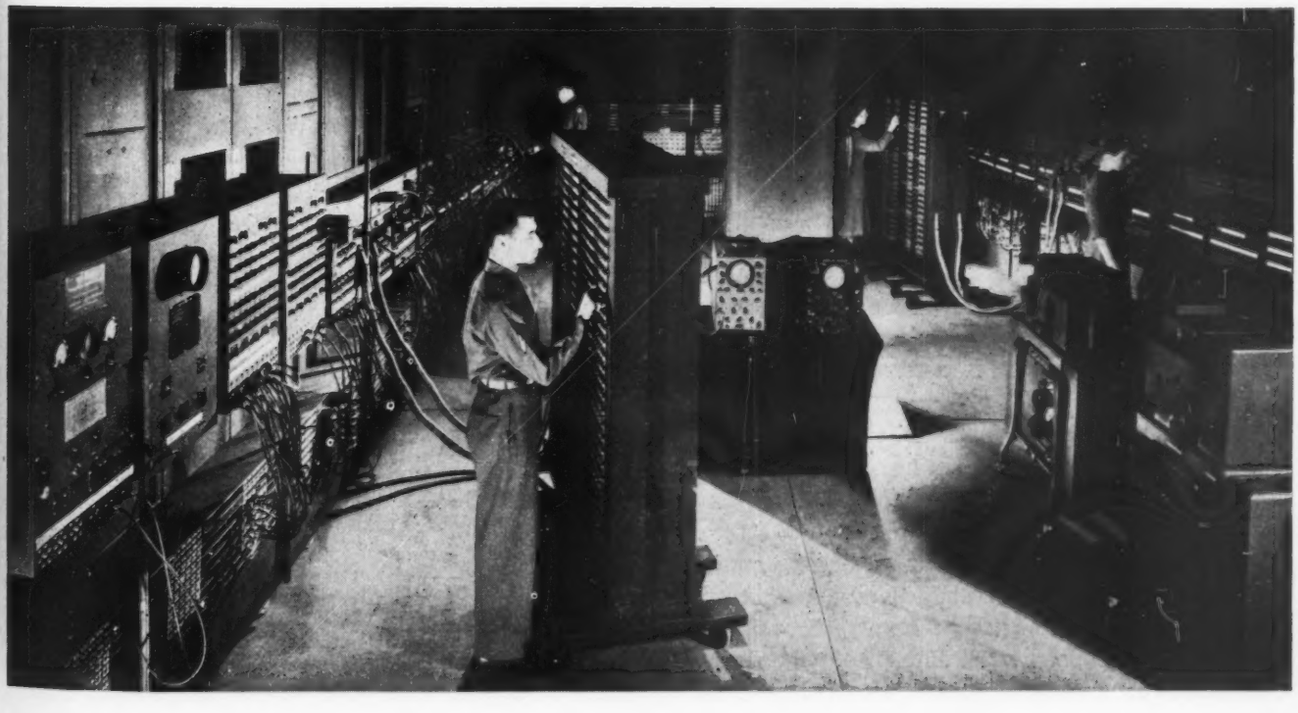
Expected to revolutionize the mathematics of engineering, the machine can handle hundreds of different factors in one computation. Much lengthy and costly design experimentation could be eliminated through its use although the machine does not remove the need for legitimate experimentation nor does it replace original human thinking. Rather it will free thought from the drudgery of lengthy computations.



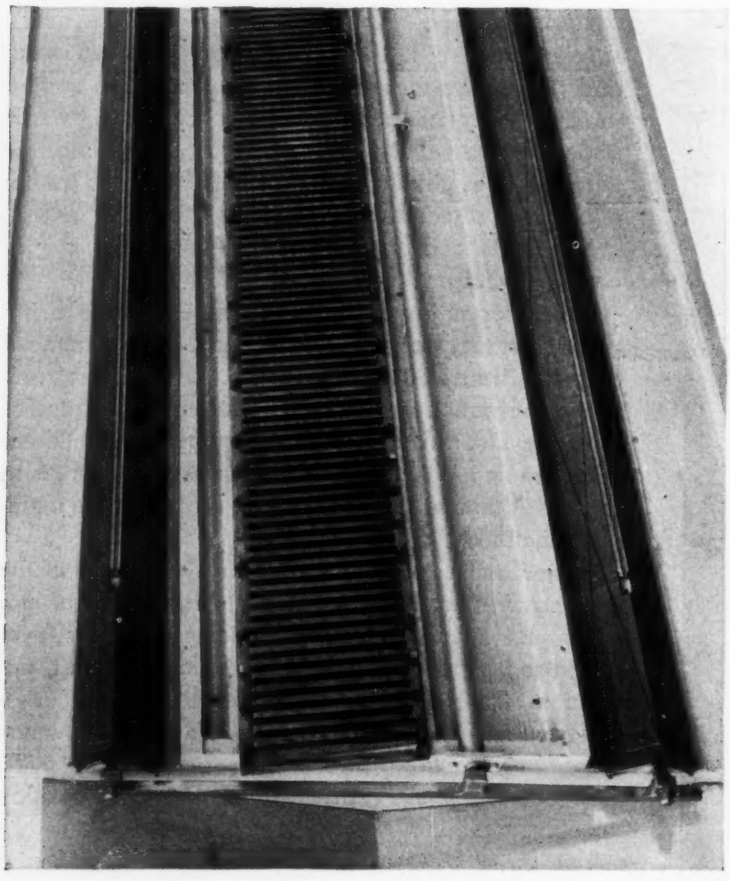
Information is fed to ENIAC from punched cards and the results therefrom are punched in a similar manner. Consuming 150 kilowatts supplied by a 3-phase regulated, 240-volt, 60-cycle power line, the machine uses 80 kw for heating the tubes, 45 kw for generating direct current voltages, 20 kw for driving a blower to cool the vacuum tubes, and 5 kw for card punching.

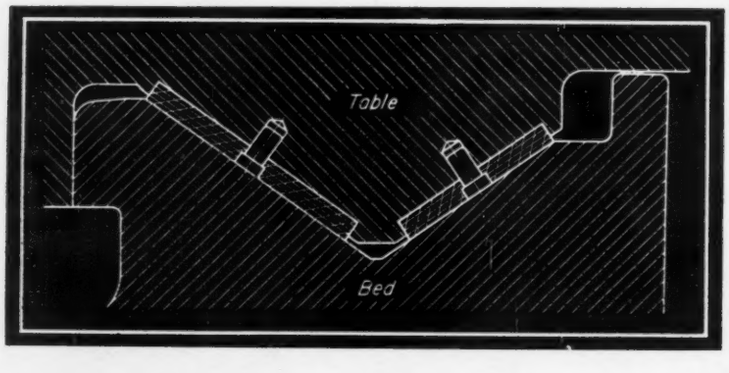
Operational units of the machine include arithmetic, memory and control elements. The arithmetic elements utilize 20 accumulators, one multiplier and one combination divider and square rooter. Internal memory is set up by circuit arrangements while external memory is supplied by the punched holes in cards. A master programmer co-ordinates the memory functions.

Major component of the control elements is the cycling unit which generates the fundamental signals. Because all units of the machine are synchronized by connections with the cycling unit, operation between pairs can be carried out simultaneously. The cycling unit contains an oscillator which generates electrical impulses at the rate of 100,000 per second. These pulses are fed into a 20-position electronic stepping switch or "counter" which enables the cycling unit to put out a program pulse at every twentieth pulse of the oscillator and a train of other fundamental pulses during the interval between program pulses. These pulses form the



basis of the programming system and set off the beginning and end of the addition cycles which are the basic arithmetical intervals of the machine.





Plastic table ways, consisting of laminated plastic plates secured to the table by laminated plastic pins as shown in the photograph and cross-sectional sketch above, protect the machine ways against cutting, scoring and wearing. Also heat insulating properties of the plastic preclude the possibility of heat transfer to the bed. Laminations in the plastic plates as well as in the pins are at right angles to the bearing surface, a matter that experience as well as theoretical considerations have proved important.

For high-speed reciprocating members where the bearing surfaces may be subjected to heavy pressures, such as in the G. A. Gray Co. planer illustrated, the heat insulating qualities of the plastic ways is of particular value. Without heat insulation even a slight temperature rise generated at the V-bearing surfaces is transmitted to the lower plate of the box section table, causing it to expand and curl up at the ends. This curling not only destroys the accuracy of the planer work, but also permits dirt to get under the ways, further increasing the danger of cutting and scoring.

Pocket-size laundry, shown below, employs an oscillating tray on its base to agitate the tub, obviating rotary seals and moving parts inside the tub. Developed by Menasco Manufacturing Co. to fill the need of apartment dwellers and others, the washer provides a convenient method of washing small amounts of clothing in a minimum of space. Tub of the machine may be removed from its tray for boiling clothes or other purposes. Fixed fins in the tub aid agitation of water and clothes to produce a gentle washing action. Drive in the base is powered by a squirrel-cage motor, developing 1/16-horsepower at 1670 rpm to operate the oscillating tray at 104 cycles per minute through a worm and gear train. Bearings are self-lubricated and the gearcase is filled with graphite grease. The entire unit weighs but 18 pounds and holds three gallons of water when the tub is two-thirds filled.

Fig. 1—Machine for testing tilting-pad thrust bearings which normally operate with hydro-dynamic lubrication. In this machine, bearings can be loaded to 12,000 psi at 100 rpm with oil temperature 250 F, conditions which cause most bearing materials to enter the partial lubrication range

By R. W. Dayton and R. E. Adams
Battelle Memorial Institute
Columbus, Ohio

Part I

Bearing Failures—Causes and Cures

AS WITH any design, simplicity and reliability are demanded in a bearing, with the emphasis on reliability. The engineer attempts, with his control over dimensions and materials, to produce a design that will operate satisfactorily in spite of adverse environmental conditions which often will occur in service. If the engineer is successful, he will produce something as good as a turbine bearing which will run for twenty years without trouble. If he is not, he may have to change bearings as frequently as a railroad does car journal bearings.

Starting point of the design is past performance in similar applications. At present, this is the only reliable source of data, because the complexity of the problem prevents any simplified tabular data from being useful. From this point on, two courses are open—to design with a large factor of safety, or to expect trouble. If overdesign is allowable, operation is generally satisfactory. On the other hand, if circumstances are such that the utmost is required, it is necessary to design, build, test, and redesign on the basis of performance, until satisfactory operation is attained. In this cut-and-try manner, the performance of aircraft bearings has been pushed up and up until it far exceeds that of any other kind of bearing.

With either approach to the design, it is essential that the engineer know the causes of, and the cures for, different types of bearing failures. It is the intention of the authors to supply this type of data. The subject of bearings is such a complex one that it is impossible to discuss it intimately in any article of reasonable length. Therefore, the intention in the present series of two articles is to discuss the performance and failure of bearings in general terms, with the view of presenting an integrated understanding of the subject. To obtain the detailed information essential for utilizing what is known of bearings, the reader is referred to the outside references listed at the end of each article.

For the purposes of this discussion, a bearing surface is defined as any pair of mating surfaces between which there is relative movement of any degree whatsoever. This will include some types of surfaces, such as press fits or clamped joints, which the engineer does not normally consider as bearing surfaces. Nevertheless, there often are tiny movements between these surfacs which cause failures like those of bearings, so a discussion of such surfaces is proper.

TYPES OF BEARING FAILURE: Bearing failures may be

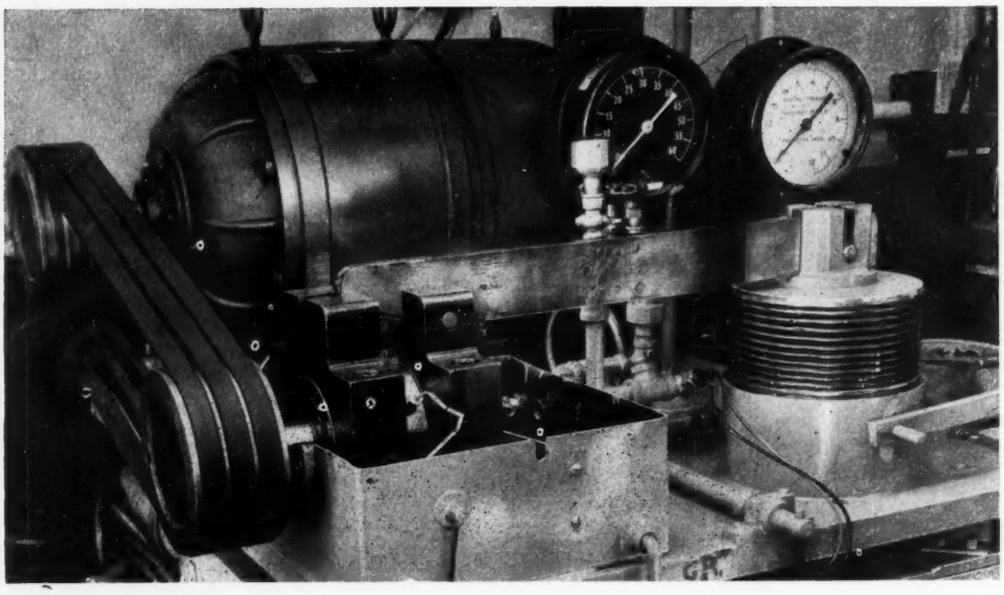


Fig. 2—Machine for testing seizure of sleeve bearings. Loads up to 12,000 psi, speeds up to 10,000 rpm and temperatures up to 300 F can be used, while grit can be fed to the bearing and the supply of lubricant interrupted

either frictional or mechanical in nature. The major types of frictional failure are wear, scoring, galling, and seizing. Mechanical failures of bearing surfaces will be mentioned only briefly.

All frictional types of bearing failures, and these are the most serious and the most difficult to control, arise because of 1, contact between rubbing surfaces and 2, relative motion. To avoid such failure it is necessary either to prevent contact (without yet defining all that is implied by contact) or to prevent motion. If either can be done, the bearing part of the problem is solved. Generally, the first is attempted; only rarely is it possible to accomplish the second. Thus, a great deal of emphasis will be placed upon methods of obtaining separation of the bearing surfaces.

TYPES OF BEARING SERVICE: The many specific types of bearing service can be grouped into three major categories, on the basis of lubrication and motion. Since it will be shown that these factors are the ones which control both performance and failure, it is a natural grouping The three categories are:

1. Hydrodynamic lubrication, continuous movement
2. Partial lubrication, continuous movement
3. Partial lubrication, reciprocating movement.

It may be noticed that rolling friction is not mentioned. This case was omitted because performance under such conditions is governed primarily by mechanical considerations which are adequately understood by the engineer.

Performance is best, and difficulties least, in the first of these three groups, because the separation of the surfaces is greatest. In the subsequent cases the mating surfaces are more nearly in contact so that failure is more apt to result. At the extreme, with small reciprocating movements, there is a scrubbing action which brings the surfaces into intimate contact and, in the worst cases, causes failure of any materials that have ever been tried.

HYDRODYNAMIC LUBRICATION: With hydrodynamic lubrication there is a relatively thick film of lubricant between the surfaces, which avoids difficulties. Bearings designed for hydrodynamic lubrication carry greater loads at higher speeds and with fewer difficulties than bearings not so designed. In any design where it is possible, hydrodynamic lubrication should, therefore, be sought.

The development of rolling-mill bearings provides a good example of the benefits of such a course. Rolling-mill brasses, the original form of bearing, are crudely designed and made, and hydrodynamic lubrication is not achieved. Wear and friction therefore, are high, and frequent replacements are required. Because of the shortcomings of metallic bearings under these conditions, nonmetallic bearings have replaced them, to some extent, with improved results (1, 2, 3) *. These are not, however, a complete solution, because they do not avoid high friction or wear; they merely reduce them. More recently, a rolling-mill bearing has been produced (4, 5) in which hydrodynamic lubrication is achieved through good design and precision manufacture, and the troubles of rolling-mill brasses thereby eliminated.

The same has been true of thrust bearings. Initially, these were plain thrust collars, in which hydrodynamic lubrication did not exist. The allowable loads were low, and performance poor. The design of the Kingsbury or Michell bearing (6) achieved hydrodynamic lubrication by using tilting pads of bearing metal which adjust their angle of tilt to the operating conditions. The tapered land thrust bearing achieves the same result by machining into the thrust surface the wedge required for hydrodynamic lubrication (7).

MECHANISM OF HYDRODYNAMIC LUBRICATION: Hydrodynamic lubrication depends on the viscosity of the lubricant, the geometry of the bearing surfaces, and on motion to support the load on a relatively thick film of oil between the bearing surfaces. The fundamental mechanism which produces hydrodynamic lubrication was explained first by Reynolds (8). Since that time, his basic theory has been applied to many specific types of bearings by others. Recent discussions of the subject, which are especially useful, are those of Hersey (9) and those of Muskat and coworkers (10). Conventional thought on the subject has been that motion of the shaft is necessary to draw oil between the surfaces but, more recently, Underwood (11)

*Numbers in parentheses refer to References listed at end of article.

has shown that variation of the direction of loading produces the same effect, and that this too was anticipated by Reynolds.

The geometry necessary for hydrodynamic lubrication is a wedge-shaped space between the bearing surfaces, into which oil is drawn. This wedge is provided by the clearance of a sleeve bearing, the tilting of a tilting-pad bearing, or the taper of a tapered land thrust bearing.

Incidentally, the load-supporting pressures which are developed in the oil film should be as extensive as possible. If oil grooves are cut through a loaded zone, they partly destroy hydrodynamic lubrication. When hydrodynamic lubrication is sought, oil grooves should be used only to distribute the oil at an unloaded point, or to obtain greater oil flow and, therefore, a better cooling, as in some cases with a circumferential oil groove(12).

Reasons for Failure: Theories of hydrodynamic lubrication apply only to the operation of such bearings, not to their failure. Theoretically there is no reason why any loads whatsoever could not be carried by any bearing materials. In the laboratory this conclusion appears to be substantiated, for no true upper limit to the load-carrying ability of a hydrodynamically lubricated bearing has ever been found. In the authors' laboratory, loads of 10,000 to 12,000 psi have been carried on bearings at speeds from 100 to 9,000 rpm, while operating in the hydrodynamic range of lubrication, *Figs.* 1 and 2. Other laboratories have reported even higher loads.

In practice, bearings cannot be depended upon to carry such loads. Thus it is evident that, although engineers may design for and hope for hydrodynamic lubrication, it does not exist at all times, and the design must take account of this fact.

Hydrodynamic lubrication is interrupted by grit, rough surface finishes, or local overloading. Momentary interruption of hydrodynamic lubrication does not cause harmful results, so long as it is immediately resumed. However, if conditions are unstable owing to high loads or speeds, a momentary interruption can lead to the type of failure known as seizure.

What appears to happen in seizure is that local heating is caused initially by contact between the bearing surfaces or by grit, and that this heating causes local expansion and still harder rubbing, until the heat generated is sufficient to destroy the bearing, or the friction becomes so high as to stop, or seize, the shaft.

Design Factors Which Influence Seizure: The kind of bearing material has a large effect on seizure, as has been discussed at length by Gillett et al.(13). With some bearing materials, the momentary contacts which are obtained between shaft and bearing result only in pushing, or wearing, the bearing metal away from the zone of high contact pressures, and the bearing heals itself and continues to operate. With others, such contact can cause a roughening or galling (welding) of the bearing surfaces that quickly destroys the bearing. In general, the soft bearing materials, babbitts, and cadmium alloys are able to operate in spite of momentary contacts, while the harder bronzes are not.

Dirt is one of the causes of seizure, and the best solution is to keep dirt out. The effect of dirt can be greatly reduced by using softer bearing materials in which the dirt can be embedded before it harms the shaft (14, 15).

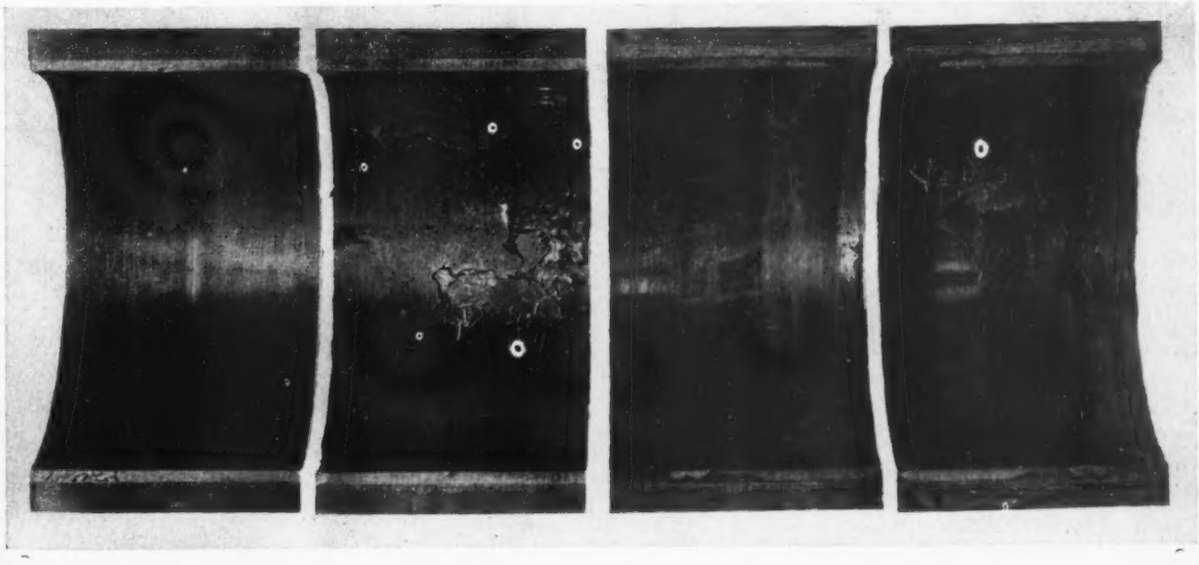
Conforming to Shaft Deflections

Uneven bearing contact, resulting from shaft deflections, can be reduced by shaping the bearing so that it conforms to the deflections of the shaft. Aircraft bearings are bored with a hyperboloidal shape to avoid this source of trouble. Another source of seizure is rough bearing surfaces, which is avoided by the obvious solution of smooth finishes.

Clearance in a sleeve bearing has a pronounced effect on seizure from any these causes. When the clearance is small, a momentary contact can easily heat the bearing surfaces so the clearance is reduced to nothing, and seizure naturally follows. On the other hand, when the clearance is too large hydrodynamic lubrication is not sufficiently effective, and wear and scoring may result. Thus, there is a proper balance of clearance for the best results.

Another aspect of clearance is that a large part of the cooling of a bearing is obtained from the flow of oil through the clearance space, which cannot be too small or the bearing will overheat. Burwell(16) discusses the question of clearance from theoretical considerations, but

Fig. 3—Babbitt bearings which failed by fatigue. Rod halves have cracked more than the cap halves



arrives at clearance values smaller than experience has shown to be desirable. This is further indication that optimum hydrodynamic lubrication is not achieved in practice.

OTHER FRICTIONAL FAILURES: Seizure can result also from a complete failure of hydrodynamic lubrication. In bearings which must start and stop under load, a hydrodynamic oil film cannot support the shaft at low speeds. There is then metallic contact and wear. Normally the rate of heat generation is too low under these conditions to cause appreciable damage, or seizure, in any single start and stop. Rather, the effect is a gradual alteration of the shape of the bearing which eventually results in a shape which will not support hydrodynamic lubrication. The small amounts of wear at each start and stop gradually wear the bearing into a valve seat, and a failure results. This effect has been observed both with partial journal bearings and with tilting-pad thrust bearings.

In some experiments made in the author's laboratory, it appeared that the nature of the bearing material had a great deal to do with this kind of failure. Harder materials, especially copper-lead, were more free of this type of failure.

One solution to this problem is to float the shaft by high-pressure oil during each start and stop. This device is used in some large electrical machinery, and has been tested in railway journal bearings by Welter and Brasch(17).

Another type of failure is erosion, which occurs occasionally with ultrahigh-speed, heavily loaded bearings. It manifests itself as pits in the bearing metal surface, which gradually enlarge until the bearing is destroyed. Little is understood of this phenomenon at the present time.

Wear of bearings, when it occurs, indicates either that there is grit present, or that lubrication is inadequate. In the first case, it is generally necessary to remove the grit, inasmuch as materials hard enough to resist abrasion cannot be used, except in isolated instances. Wear due to the second cause is more properly discussed in the section on partial lubrication.

Preventing Mechanical Failure

MECHANICAL FAILURES OF BEARINGS: Mechanical failure of the soft bearing materials sometimes occurs. Squashing or pounding out occurs infrequently, generally only with thick, unbonded babbitt bearings. In other cases the strength of the bearing exceeds the loads, and there is no such difficulty.

Fatigue failure, however, is a more serious problem. This may occur with any repeatedly loaded bearing, such as a connecting-rod bearing, *Fig.* 3. The automotive industry has combated this problem partly by decreasing the thickness of the babbitt(13, 18), which improves the life by making a stronger structure. The aircraft industry, too, has used this device, and the best aircraft bearing available at the present time is a lead-indium or a lead-tin alloy, used in a very thin layer backed up by silver and steel. The composite bearing is able to carry enormous loads without difficulty.

If thin coatings cannot be used, it is necessary to employ stronger materials. The available materials, listed in the order of increasing strength, are:

1. Babbitt, lead and tin base
2. Cadmium-base alloys
3. Copper-lead
4. Silver
5. Bronze.

Aluminum bearings, which are now finding some use here and abroad, have properties which are in the range of those of copper-lead alloys.

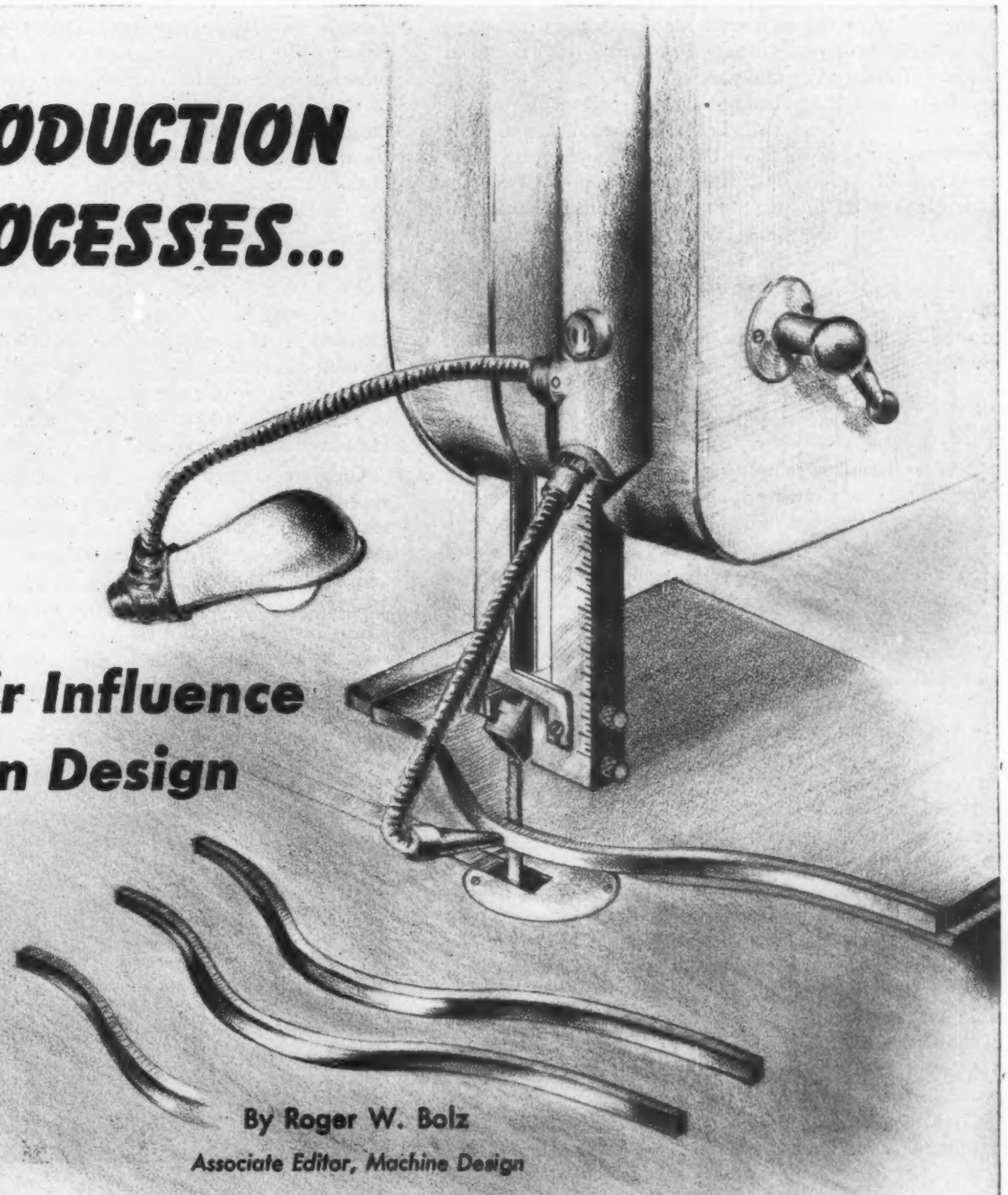
The use of babbitt in connecting-rod bearings of radial aircraft engines was abandoned long ago in this country, because of insufficient fatigue strength. However, babbitt is still being used successfully in some British radial aircraft engines, even though the bearing loads are about the same as in our engines. The difference in practice which explains this situation is that, in the British engines, the babbitt is used on the shaft, where the loads it carries are practically constant, rather than varying. Since there are no repeated loads, there are no fatigue failures. Recent tests by a British investigator(19) have shown that in automobile engines, too, better bearing life is obtained by applying the bearing metal to the shaft rather than to the rod. In the concluding article the performance of bearings having partial lubrication will be discussed. Both continuous movement and reciprocating movement will be covered, with emphasis on the prevention of failure due to galling and wear.

REFERENCES

1. A. S. T. Thomson—"Film and Friction Measurements on Journal Bearings, Including Fabric Types", *General Discussion on Lubrication and Lubricants*, A.S.M.E., Part I, Page 323.
2. E. Watson Smyth—"Some Applications of Fabric Bearings", *General Discussion on Lubrication and Lubricants*, A.S.M.E., Part I, Page 277.
3. H. Rochester—"Fabric Bearings for Rolling Mills", *General Discussion on Lubrication and Lubricants*, A.S.M.E., Part I, Page 260.
4. H. H. Wood—"Lubrication of Morgoil Roll Neck Bearings", *Iron and Steel Engineer*, Vol. 14, Nov., 1937, Pages 36-40.
5. W. Trinks—"Flood Lubricated Bearings", *Blast Furnace and Steel Plant*, Vol. 22, June, 1934, Page 343; July, 1934, Page 403.
6. J. Hamilton Gibson—"Pivoted Pad Thrust Bearings", *General Discussion on Lubrication and Lubricants*, A.S.M.E., Part I, Page 107.
7. F. C. Linn and R. Sheppard—"The Tapered Land Thrust Bearing", *General Discussion on Lubrication and Lubricants*, A.S.M.E., Part I, Page 171.
8. O. Reynolds—"Theory of Lubrication and Its Application to Mr. Beauchamp Tower's Experiments", *Phil. Trans. Roy. Soc.*, Part I, Vol. 177, Pages 157-234, 1886.
9. M. D. Hersey—*Theory of Lubrication*, John Wiley & Sons, New York, 1936.
10. M. Muskat and Co-workers—"The Theory of Thick Film Lubrication of a Complete Journal Bearing of Finite Length", *J. Applied Phy.*, Vol. 9, June, 1938, Page 393; "Experimental Friction Coefficients for Thick Film Lubrication of Complete Journal Bearings", *J. Applied Phy.*, Vol. 9, August, 1938, Page 539; "Theory of Thick Film Lubrication of a Complete Journal Bearing of Finite Length With Arbitrary Positions of the Lubrication Source", *J. Applied Phy.*, Vol. 10, Jan., 1939, Page 47; "The Experimental Variation of the Coefficient of Friction With the Strength of the Lubricant Source for a Complete Journal Bearing, *J. Applied Phy.*, Vol. 10, May, 1939, Page 327; "The Theory of Thick Film Lubrication of Flooded Journal Bearings and Bearings With Circumferential Grooves", *J. Applied Phy.*, Vol. 10, June, 1939, Page 399; "The Lubrication of Plane Sliders of Finite Width", *J. Applied Phy., Vol.* 11, March, 1940, Page 208; "Lubrication of Plan Sliders", *J. Applied Phy.*, Vol. 11, Aug., 1940, Page 541; "Effect of Pressure Variation of Viscosity on the Lubrication of Plane Sliders", *J. Applied Phy.*, Vol. 11, Nov., 1940, Page 739.
11. Arthur F. Underwood—"Rotating Load Bearings", *Automotive and Aviation Industries*, Vol. 92, June 1, 1945, Page 26.
12. R. J. S. Pigott—"Bearings and Lubrication"—*Mech. Eng.*, Vol. 64, April, 1942, Page 259.
13. H. W. Gillett, H. W. Russell, and R. W. Dayton—"Bearing Metals from the Point of View of Strategic Materials"—*Metals and Alloys*, Sept., Pages 274-283; Oct., Pages 455-463; Nov., Pages 629-639; and Dec., Pages 749-758.
14. H. C. Mougey—"The Newer Bearing Materials and Their Lubrication"—*Ind. and Engineering Chemistry*, News Edition, Nov. 10, 1936, Pages 425-8.
15. S. Epstein and R. C. Hess—"Bearing Metals in the Steel Industry"—*Iron and Steel Engr.*, Vol. 21, Aug., 1944, Pages 83-93.
16. J. T. Burwell—"The Effect of Diametral Clearance on the Load Capacity of a Journal Bearing", *A.S.M.E. Trans.*, Vol. 64, 1942, Pages 457-61.
17. G. Welter and W. Brasch—"Tests of Plain Bearings With a New Method of Lubrication Under Very High Pressure", *General Discussion on Lubrication and Lubricants*, A.S.M.E., Part I, Pages 330-335.
18. Arthur F. Underwood—"Automotive Bearing Materials and Their Application", *S.A E. Jour.*, Vol. 43, 1938, Pages 385-92.
19. Tom Brown—"Effect of Whitemetalling Crankshaft Journals", *J. Inst. Automobile Engineer*, Feb., 1944, Pages 24-27; Vol. 12, 1944, Page 205.

PRODUCTION PROCESSES...

Their Influence On Design



By Roger W. Bolz

Associate Editor, Machine Design

Part XI–Contour Sawing

MOST identical machines on the scene today are looked upon as products of the various so-called mass production processes. However, survey has shown that but 20 per cent of those manufactured ever exceed a production figure of 10,000. Thus, to achieve in such medium and low-production items as street cars, printing presses, aircraft, scientific equipment, machine tools, and special machines the same qualities of low cost, design versatility and inter-

changeability as obtained with high-production processes, contour machining or sawing often can be resorted to in the manufacture of certain parts.

With plain contour band sawing, it is possible to produce at low cost from a single block of steel high-strength complex parts, *Fig.* 1, that would require otherwise much more expensive processing. Likewise, it is possible to produce blank parts in quantities from one to five hundred pieces by this process, eliminating the usual blanking die costs for such parts, *Fig.* 2. Sheets are compressed and welded together to simplify the sawing process. Intricate shapes are cut with the same ease as simple ones, chief controlling factors being the length of cut and material used.

Eliminates Expensive Machining

Beside providing sample forgings, substitutes for forgings, etc., contour sawing simplifies in many cases the design of large forgings. Simplified blank forgings can be rapidly cut to proper contours, making possible the use of such a blank for a number of basically similar parts. In aircraft work, for instance, right and left-hand forked fittings are contour sawed from the same forged blank, not only eliminating an extra forging die but also the need for an expensive straddle-milling job.

Without doubt, on such jobs, contour sawing is one of the fastest methods known for removing bulk metal and leaving a relatively good surface. Subsequent finish machining time is thus reduced to a minimum, *Fig.* 3.

In addition to the wide variety of standard types of two and three-wheel band machines available for normal-velocity work, open-end band machines are available. These carry the toothed band helically wound on a drum or reel and require no blade welding or brazing. Up to 2.5 minutes of sawing can be had before the end of the band is reached and rewinding to start position is necessary. Extremely accurate and delicate contours are possible with such machines with, however, the disadvantage of the short duration of cut.

During recent years, the contour sawing field has been broadened to a considerable extent by the perfection of high-velocity sawing, usually referred to as friction band sawing, *Fig.* 4. Basically a frictional melting or burning process, this superhigh-speed method of contour cutting has stepped up production of extremely hard and often otherwise unmachinable materials used in the manufacture of ships, planes, automotive vehicles, railway equipment, farm machinery, tanks, etc. Primary difference between the conventional and friction band sawing lies principally in the surface speed of the band and feed pressure. While the conventional sharp saw tooth cuts chips and carries them out of the kerf oftentimes under a coolant, the dull dry friction saw develops sufficient frictional heat at the tooth contact to bring a material instantaneously to melting temperature for easy removal. A fine jet of compressed air directed upon the blade often assists in the burning operation, *Fig.* 5. This burning operation extends the range of materials available for contour sawing to embrace an unusual variety, including glass, plastics, Hastelloy, armor plate, etc.

Ordinary contour sawing is normally accomplished at speeds from 50 to 3000 fpm depending upon the type of material and the thickness. High-velocity sawing, however, begins where conventional metal sawing leaves off and today machines are in use which deliver as high as 16,200 fpm. Blades with either raker or wave-set teeth are used with a temper slightly softer than that for wood saws, to avoid premature fatigue. For material up to ⅛-inch in thickness, 18 teeth per inch are used; from ⅛ to ¼-inch thicknesses, 14 teeth per inch; and from ¼-inch on up, 10 teeth per inch are used. These blades are the low-

Fig. 1 — Above — Production of such parts as this locomotive poppet valve lever arm is simplified by contour sawing

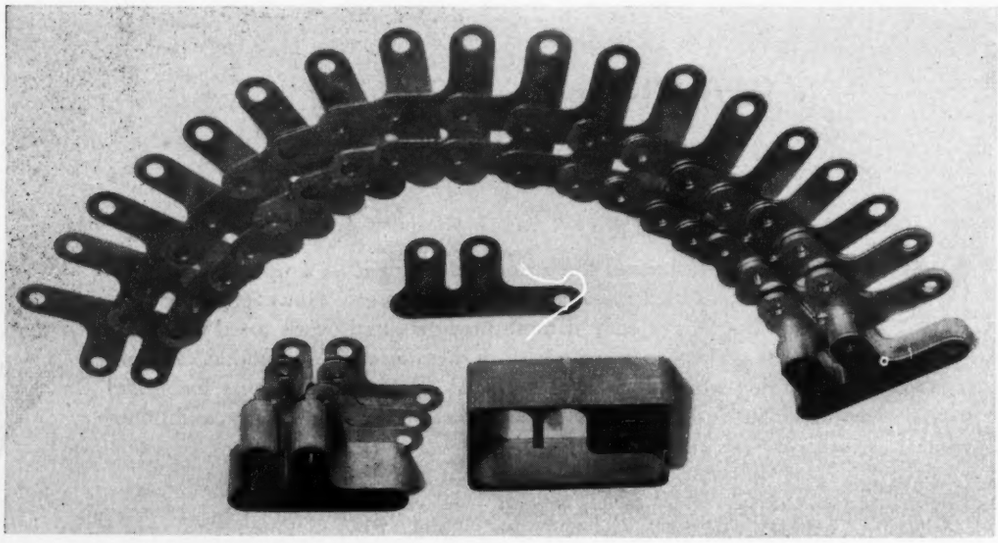


Fig. 2—Left—Contour sawing stacked sheets at conventional speeds, produced these 50 identical electric switch parts simultaneously

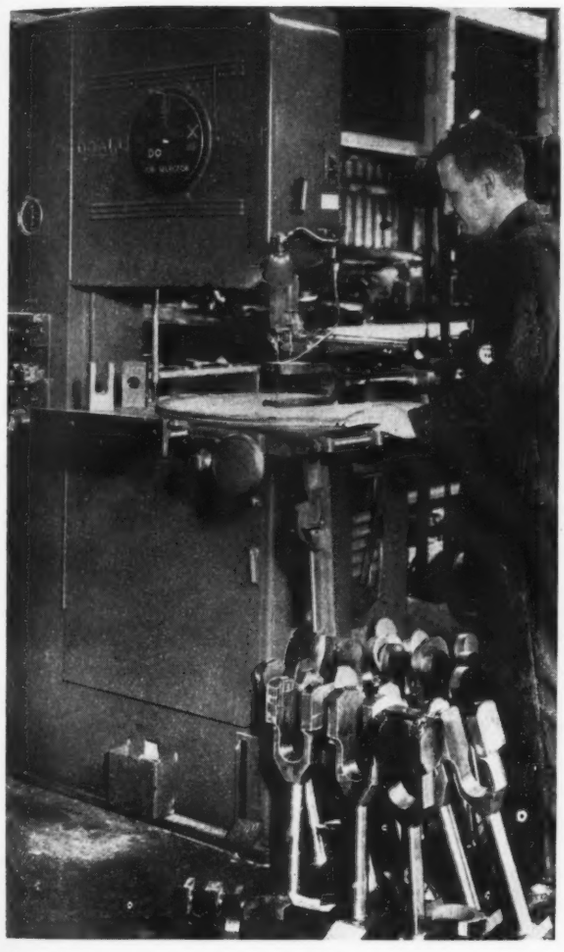


Fig. 3—Above—Sawing reduced machining time for diesel engine connecting rods from 8 hours to less than 1 hour per rod

Fig. 4—Above—Friction sawing extends the scope of contour work to many materials not possible to machine by other methods

est in cost and in production may last up to 40 hours.

Most present-day friction sawing is performed on parts made from materials up to ½-inch in thickness. Light gages of ferrous or hard materials (up to ⅛-inch) are cut using a band speed from 3000 to 5000 fpm. This speed increases to as high as 10,000 to 12,000 fpm with thicknesses up to ¼-inch. Using speeds around 15,000 fpm, up to ½-inch sections can be easily cut. Blade speed, roughly speaking, will be determined by the melting point of the material to be cut—hardness actually helps a little. Some authorities place the limit on thickness for such materials at ½-inch. However, much heavier plate has been cut and in one instance at least, using a speed of 16,200 fpm, armor plate 2 inches thick has been easily handled. These heavier gages, and especially those over 1-inch, often require a constant power feed to assure sufficient friction and in such cases torch cutting—if it can be used —is usually more economical. Nonferrous materials, on the other hand, require no special techniques and can be cut, though finish may be poor, in sections up to capacity of the machine in use.

High Linear Speed of Cutting

Friction-sawing cutting rates are considerably higher with materials up to 1-inch in thickness than rates obtained with various other methods. Speeds attained in production run from 20 inches per minute with ½-inch

Fig. 5—Below—Typical cuttings resulting from friction sawing armor plate at 12,000 fpm with a 14-pitch blade

thick materials to as high as 120 feet per minute on 1/16-inch thicknesses, *Fig.* 6. Armor plate ¾-inch thick has been cut at approximately 14 inches per minute and Hastelloy at 3 inches per minute, using a 1-inch wide, 10-pitch blade operating at 15,000 fpm. Using a ½-inch wide, 14-pitch blade operating at 12,000 fpm, 5/16-inch SAE 4140 chrome-molybdenum steel plate has been cut in aircraft production work at 20 inches per minute, 5/16-inch armor plate at 12 inches per minute, ⅛-inch stainless at 80, ¼-inch Tantung C at 3, ½-inch Delloy tool at 1, etc. Cutting speed, therefore, is not entirely dependent upon thickness of material, but to some extent upon the material hardness—the harder tempers having a higher coefficient of friction are heated more easily and cut more quickly.

In addition to speed, there is the advantage of negligible down drag on the part being cut and ability to cut light-gage sheet metal on the scarf with wide angles in either direction. This particularly suits friction sawing for odd contour trimming operations such as in the aircraft industry where it is used almost solely for trimming light gage formed work of aluminum alloys and steels.

TABLE I

Saw Width Effect on Gage and Radius of Turn

Saw Width	Maximum Material Gage	Minimum Radius
1/16	Under 1/16	1/16
3/32	"	1/8
1/8	"	7/32
3/16	1/16	3/8
1/4	1/8	5/8
5/16	3/16	7/8
3/8	1/4	1
1/2	3/8	3
5/8	1/2	5
3/4	3/4	7
1	1	9

NOTE: Maximum material gage indicates a safe limit for production considerations primarily in friction sawing hard steels, high-tensile steel alloys, armor plate, and other extremely hard materials to assure sufficient blade strength for maximum feed. Materials too thick to bring to melting temperature at available blade speeds can often be cut by using a "rocking" technique.

Minimum radii indicated hold for any thickness of nonferrous or softer materials conventional or friction sawed. Radii can in some cases be reduced further by careful handling. Use of the largest radius possible naturally allows widest blade with maximum linear cutting speed.

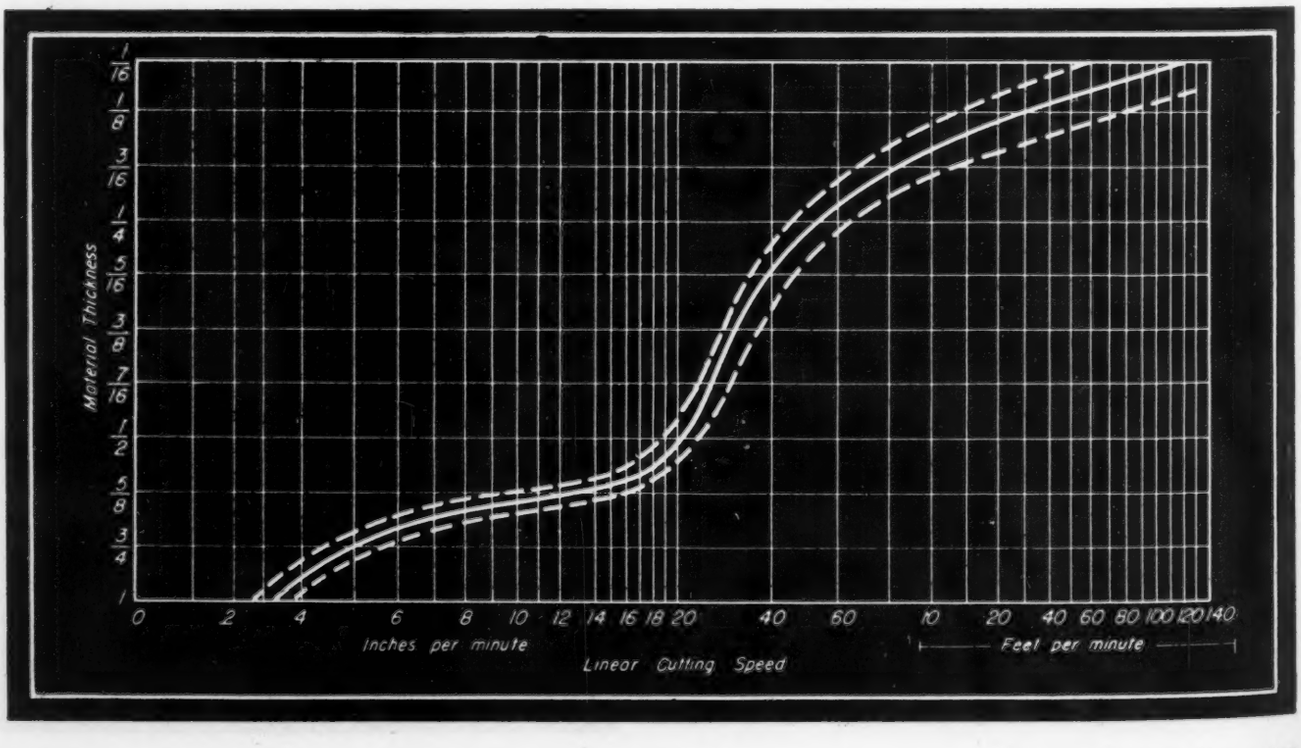
DESIGN: Although both the conventional and friction contour sawing processes are limited primarily to the production of surfaces generated by a straight line, the latitude in design variations is extremely wide. Many parts required in only nominal quantities can be produced by this method to serve both satisfactorily and economically in place of forgings. This is especially true in aircraft work where design changes rapidly and on experimental machines where time is a factor. The unit shown in *Fig.* 7 shows such a part and indicates the design possibilities.

Heavy Cuts Require Wide Blades, Large Radii

Except in the case of materials which cannot be sawed rapidly with a satisfactory finish at conventional speeds, ordinary contour sawing methods are indicated for parts over 1 inch in thickness. Generally speaking, however, the heavier the section to be cut, the wider the blade must be to maintain the maximum economical linear feed. Where intricate curves must be produced on gages heavier than those indicated in TABLE I, some sacrifice in output speed results. Open-end automatic rewind saws operating at speeds up to 180 fpm can traverse the most exacting and delicate contours using the narrowest width blades. Using friction cutting speeds, a blade of 3/16-inch width will produce almost equally exacting results in most plastics.

Typical aircraft parts which have been friction sawed in production from SAE 4140 high-tensile steel are shown in *Fig.* 8. Smaller radii than those shown in TABLE I were

Fig. 6—Below—Conservative estimate of linear cutting speeds in friction sawing carbon steels



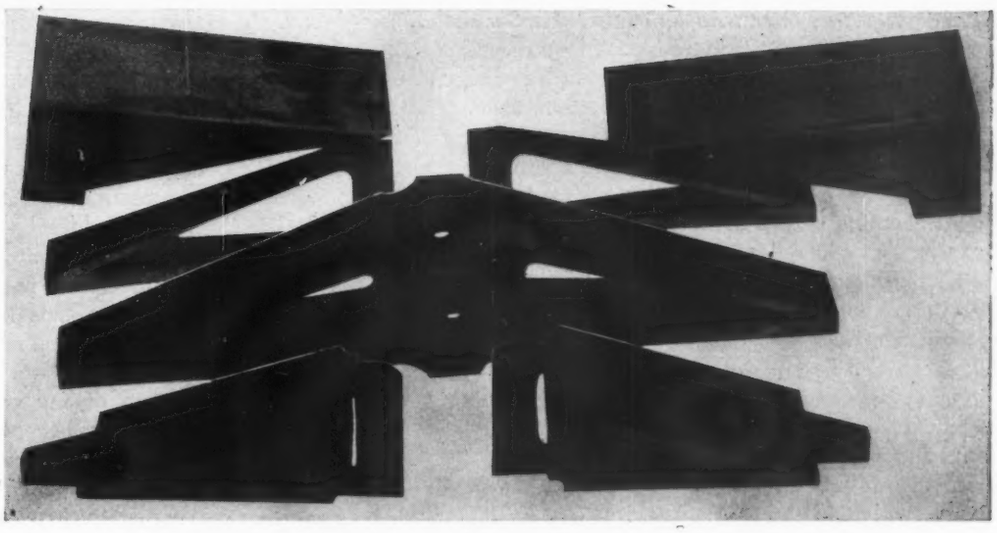


Fig. 7—Right—Substitute for a forging sawed at conventional speeds from a solid bar of high-tensile alloy steel

called for on these parts but nevertheless in every case output per hour was about six times that possible at conventional speeds.

In the manufacture of motor mounts and other parts to be fabricated from tubing, contour sawing has been found most economical and in certain cases shapes can be cut that would be impractical by any other means. Friction cutting of tubing is somewhat restricted by present equipment as is sheet and plate material. For most practical purposes maximum limitation is restricted to about one inch of tooth contact where the blade momentarily breaks through the inner tube wall. Thus if tube diameter is large, the wall section must be thin and conversely if the diameter is small, wall section can be heavier.

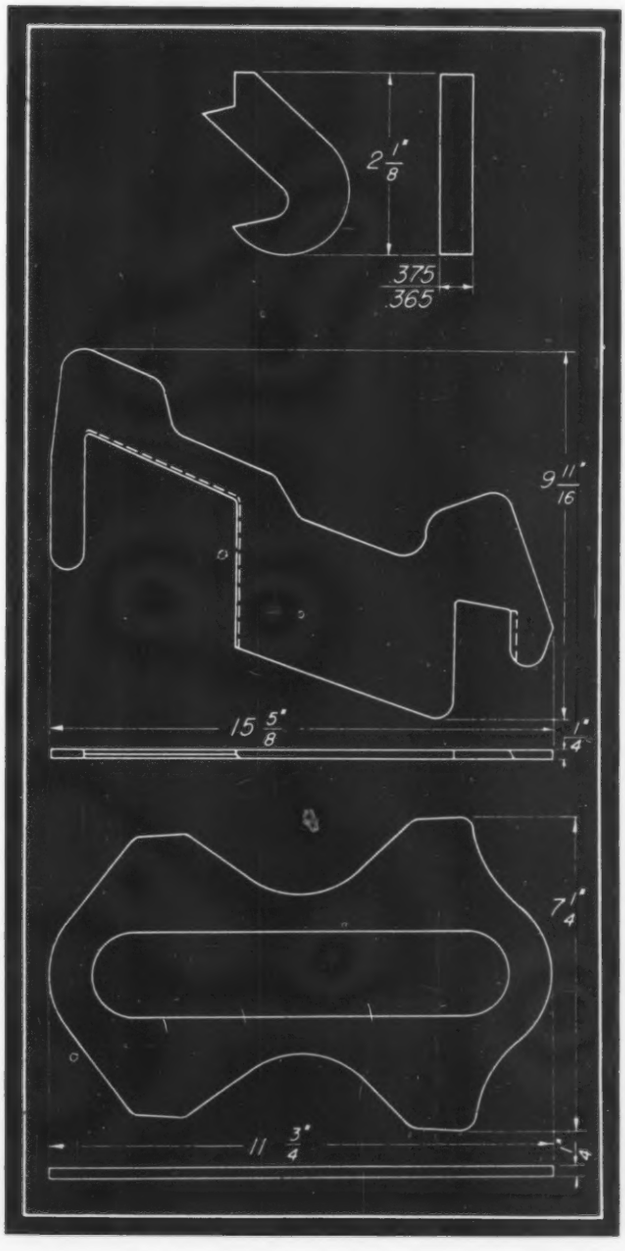
Automatic Jobs Restricted

Shapes produced on automatic contour friction saws with hydraulic feed as shown in the head illustration, are duplicated by means of a templet. Forms having radii not less than 12 inches can be cut fast, clean and square with the entire cut completely automatic.

MATERIALS: Most of the nonferrous materials such as brass, bronze, aluminum, magnesium, copper, kirksite, plastics, etc., can be handled with satisfactory feeds at conventional saw speeds. Certain types of highly accurate work such as is carried out in the making of dies also can be handled most easily at conventional speeds. However, for production-quality work, most steels and steel alloys can be handled at extremely high feeds and with relative ease by high-velocity friction sawing with a satisfactory finish. In fact, many metals that are difficult or impossible to saw by conventional methods—armor plate, stainless, etc.—can also be cut with ease. Only those materials having extremely high melting points are exempted as are those with extremely low melting points which are prone to charring or excessive stickiness. Extremely brittle materials are as a rule not well suited—they usually shatter.

Aluminum in other than sheet form is not considered to be a suitable material for friction sawing. Finish is poor and welding or gumming sometimes causes difficulty with the heavier sections. Speeds around 3000 fpm produce optimum results in such cases. However, trimming

Fig. 8—Below—Typical aircraft parts friction sawed from SAE 4140 high-tensile alloy steel. Output increased as much as 6 times over previous methods



formed shapes of aluminum and other sheet materials is done extensively in aircraft production by means of friction sawing and lack of down drag as well as free cutting on such formed material makes this method ideal.

According to certain aircraft manufacturers, most plastics can be easily handled by first dulling the saw teeth by cutting up several old files or other hard material and then smoothing with a fine hard stone. Plastics can then be cut as fast as the hand can feed material into the blade. Better finish is obtained with plastics than other materials. Certain of the newer plastics which cannot be cut with a carbide saw and those having glass, mica, and similar fillers can also be cut easily.

Plate glass can be cut, but chipping and cracking often result. Heavy insulating glass block can be cut satisfactorily and pyrex glass will finish with practically no chipping. However, the amount of polishing necessary if a smooth edge is desirable is excessive in comparison to other methods.

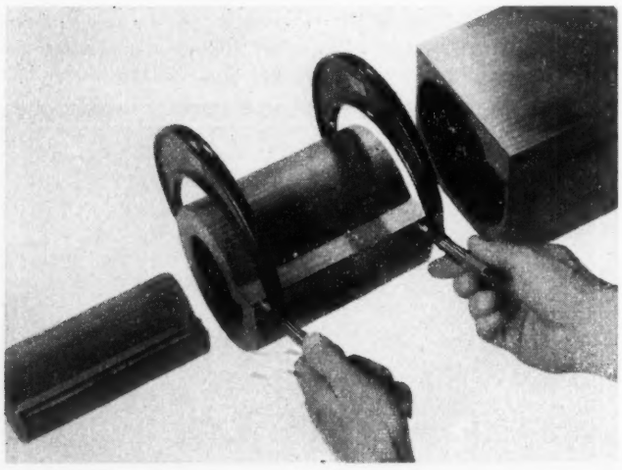


Fig. 9—Above—Cam shape cut at conventional speeds shows a variation from end to end of only 0.002-inch

Fig. 10—Below—Chrome-molybdenum forging regularly run in production showing typical surface finish left on friction-sawed surface

Except in cases where heat penetration would be detrimental, with parts over one inch in section thickness, torch cutting is the most economical method for reproducing contoured parts from ferrous materials. On materials such as Hastelloy, subsequent machining is simplified since heat of friction sawing seldom penetrates to a depth greater than 0.002-inch. On certain alloys and high carbon steels this hardening may run to as much as 0.005-inch in depth. It is good practice to anneal friction-sawed parts prior to profile milling or otherwise machining, in order to increase tool or cutter life.

Ferrous castings of any variety may be friction sawed efficiently. Section thicknesses conform to similar restrictions as those shown in TABLE I.

TOLERANCES: Conventional contour sawing produces parts with comparatively clean, uniform surfaces, requiring but little finishing. Although feed is slow, results obtained on highly exacting contours are exceptional, especially on extremely intricate designs produced with open-end machines which obviate the welded blade joint. A layout line can be followed within 0.008 to 0.010-inch without much trouble. By the use of a magnifying arrangement, a scribed line can often be followed within 0.003-inch. Vertical accuracy with conventional sawing—important especially on stack cutting—is excellent. The 6-inch high cam shape shown in *Fig.* 9 cut by this method showed a variation from one end to the other of only 0.002-inch.

Finishing Requirements Light

In most cases of contour sawing work, the parts are finished finally by some other means such as milling, filing, grinding, or polishing. The surface finish in low-speed sawing may vary from pronounced shallow scratches to scarcely visible ones. In general, the surface left by the friction saw is comparable, *Fig.* 10, but in some cases is smoother and, with plastics, almost polished.

Tolerances maintained in friction sawing are about the same as those for general low-speed work. A good operator in production can hold the contour of a part to within 0.007 to 0.010-inch of the scribed line. In fact, on special jobs, operators have been able to halve this figure, holding to around 0.010-inch overall variation, as against a normal 0.014 to 0.020-inch. Vertical runout is comparable to that of low-speed sawing.

A burr forms on the underside of the cut in friction sawing steels and some other metals, especially on the softer ones. The burr is not a problem and can be removed easily by filing. Burrs produced are thin and may project 1/32 to 1/16-inch below the cut. Also, with parts from softer materials which produce the larger burrs, cuts are made over a bar support which raises the plate from the machine table, both facilitating manipulation and preventing rapid dispersion of the heat created in friction sawing.

Collaboration of the following organizations in the preparation of this article is acknowledged with appreciation:

Bell Aircraft Corp. ..... Buffalo
The DoAll Co. (*Figs.* 1, 2, 3, 6, 7, and 9) ..... Minneapolis
Eastern Aircraft Div., General Motors Corp.
(*Figs.* 4, 5, 8, and 10) ..... Linden, N. J.
Grob Brothers ..... Grafton, Wis.
The Tannewitz Works ..... Grand Rapids, Mich.

Fig. 1—Right—Natural scales for the first, second and third powers as well as a logarithmic scale

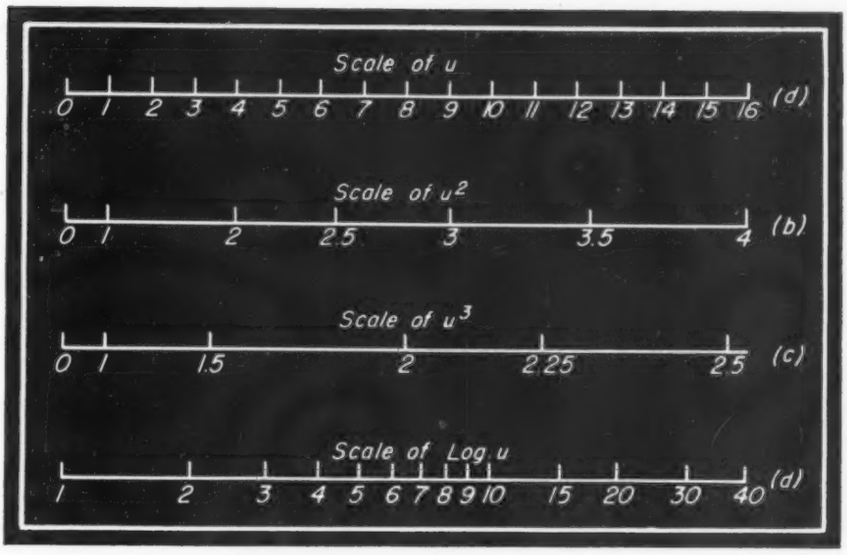
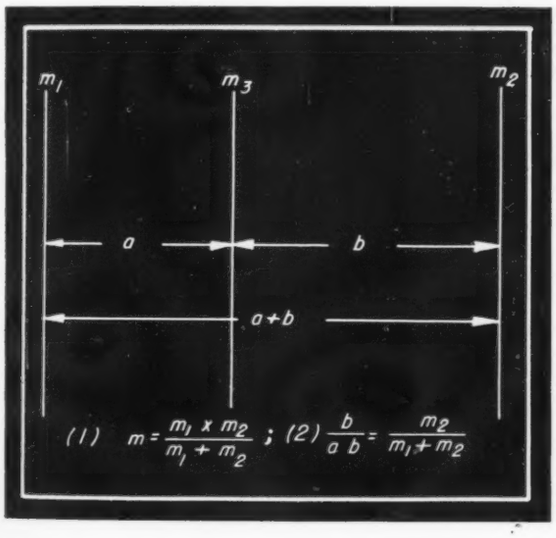


Fig. 2 — Below—Relationships between moduli and distances between nomograph scales



Simplifying the Construction of Nomographs

By E. B. Steinberg
Engineer, Electrical Dept.
Allis-Chalmers Mfg. Co., Milwaukee

NOMOGRAPHS, having found applications in many fields of engineering, have become established as time-saving mathematical tools in mass production operations and in various fields of design requiring repeated calculations with one and the same formula. Although much has been written in explanation of the use of these engineering aids, the universal adoption of nomographs is still hampered by the widespread misbelief that a knowledge of higher mathematics is necessary to prepare them.

Actually, most problems can be solved with the help of a simple nomograph consisting of parallel straight lines, one for each variable. Such a nomograph can be designed easily without knowledge of determinants. The clearness and readability of a nomograph depends largely on the setting-out of the scales which are spaced from each other in a definite relationship to make the nomograph or alignment chart. Consequently, sufficient time should be spent in studying the layout.

In *Fig.* 1, scales *a*, *b* and *c* are called "natural scales". Scale *a* represents the function of $y = u$, scale *b* is the function of $y = u^2$ and scale *c* the function of $y = u^3$. Scale *d* is a logarithmic scale: $y = log\ u$. The scale to be used depends upon the mathematical relationship between the members of the function. Another basic concept is the "scale factor", also called "modulus", which is the length of scale divided by the range of function. These are chosen by the designer of the nomograph, who considers the needed range of the function and size of paper.

For example the modulus may be found as follows: If it is desired to have a range of 15 to 45 for function $y = x$ and the length of scale is to be 3 inches then

$$m = \frac{3}{45-15} = \frac{3}{30} = 0.10$$

Also, as another example, a range of function 10 to 250 for function $y = u^{0.85}$ is desired to have a scale length of 5 inches. It is obvious that a logarithmic scale is most suitable. The modulus then becomes

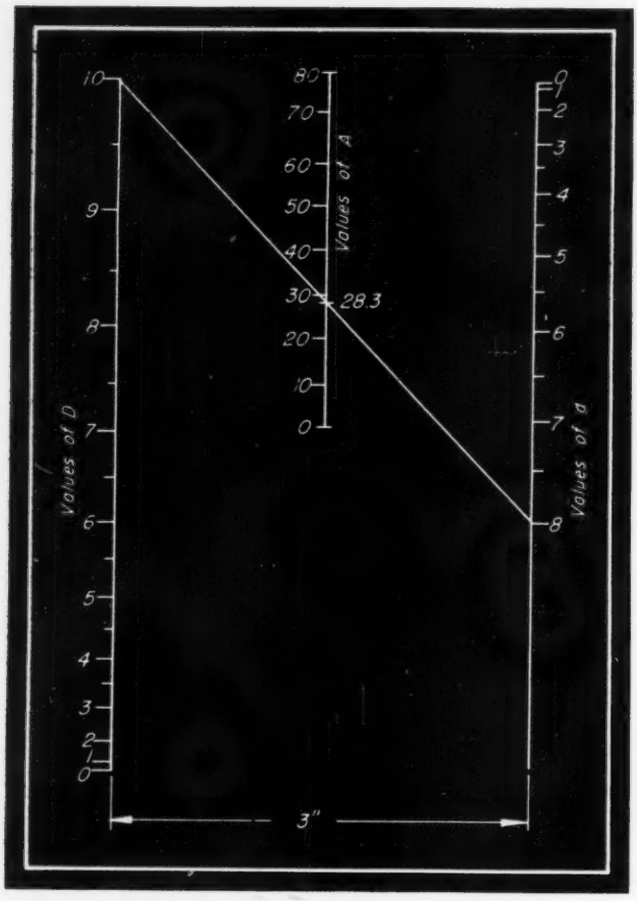


Fig. 3—Above—Simple nomograph constructed for area, A, of an annulus having outside diameter D and inside diameter d

$$m=\frac{5}{0.85(\log 250-\log 10)}=\frac{5}{0.85(1.3979)}=4.20$$

If there are any constants in the formula or equation, they should be combined into one constant which will be assigned to one of the variables.

Procedure for constructing a nomograph with parallel straight lines can be observed in the following examples where charts are constructed to give the area of an annulus and the synchronous speed of motors, respectively.

CHART GIVING AREA OF ANNULUS: The formula for the area of an annulus is $A = \pi (D^2-d^2)/4$. It is desired to have scale D from 0 to 10 and d from 0 to 8. For convenience in the subsequent operations, the equation for the annulus may be transformed into the form

$$\frac{A}{0.7854}=D^2-d^2$$

Assuming modulus m_1 for D, m_2 for d and m_3 for A with the D scale to be 5 inches,

$$m_1=\frac{5}{100}=.05$$

Assuming m_2 to be of the same modulus, m_3 is given by the relation of Equation 1 in *Fig. 2*:

$$m_3=\frac{0.05\times 0.05}{0.10}=\frac{0.0025}{0.10}=0.025$$

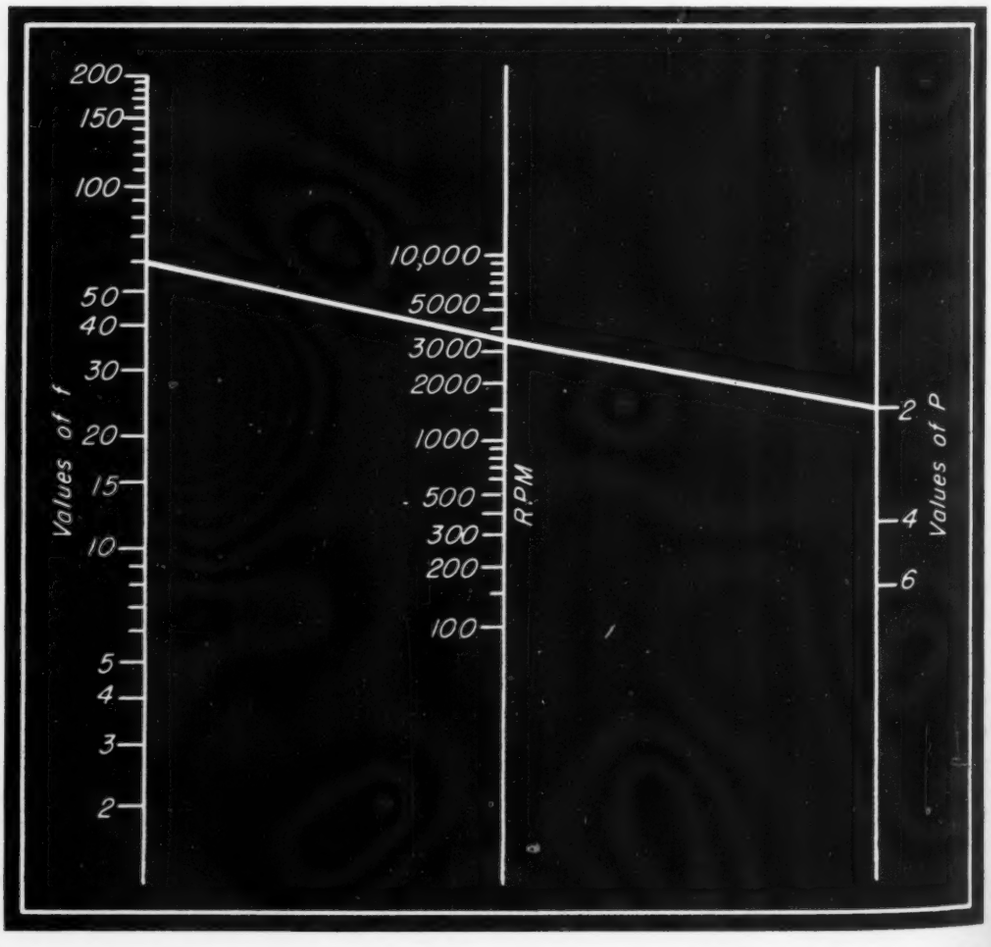


Fig. 4 — Right — Synchronous-speed chart for an induction motor requires logarithmic scales

Choosing the distance between D and d scale as 3 inches and referring to Equation 2 in *Fig.* 2,

$$\frac{b}{3}=\frac{0.05}{0.10}$$

$$b=\frac{0.15}{0.10}=1.5$$

Then scales for D and d are designed. Scale d is to be reversed, however, because its values are subtracted from D. In order to locate one reference point on A scale, one example is calculated

$$D=10, d=8$$

$$A=(100-64)0.7854=28.3$$

Drawing a line from 10 on D scale to 8 on d scale intersects A scale at the point 28.3. The A scale can now be completed as follows: The function of A is $A/0.7854$ or $1.273A$. This value times m_3 equals $1.273A \times 0.025 = 0.0318A$. Therefore, 10 units of $A = 0.318$ inches, 100 units of $A = 3.18$ inches and so on. A table can be calculated and the values plotted on the A-line accordingly. The finished nomograph is shown in *Fig.* 3.

CHART FOR SYNCHRONOUS SPEED OF MOTORS: The following example involves logarithmic scales. Required: A nomograph giving the synchronous speed of an induction motor.

$$N=\frac{60f}{p/2} \text{ or } \frac{120f}{p}$$

where N = Number of revolutions per minute, f = frequency and p = number of poles. Writing this equation in logarithmic form, a simple nomograph is obtained

$$Log\frac{N}{120}=log\, f-log\, p$$

It is desired for f to range from 0 to 200, assigning m_1; p to range from 2 to 6, assigning m_2; f scale to be 4.6 inches; and p scale to be 0.954 inches. Then

$$m_1=\frac{4.6}{log\ (200-D)}=\frac{4.6}{2.301}=2$$

$$m_2=\frac{.954}{log\ (6-2)}=\frac{.954}{.4772}=2$$

$$m_3=\frac{4}{4}=1$$

Choosing $a + b$ to be 4 inches,

$$\frac{b}{a+b}=\frac{m_2}{m_1+m_2}$$

$$b=\frac{(a+b)m_2}{m_1+m_2}=\frac{4\times2}{4}=2 \text{ inches}$$

Two lines are drawn 4 inches apart. The left line should be the f scale, and is divided as a log scale from 1 to 200 with a modulus of 2. The right scale is the p function with values ranging from 6 to 2, and is reversed since the log function is minus.

Next, one point is calculated on the N scale by using the original formula where $f = 60$, $p = 2$, and $N = 3600$. Drawing a line through the respective points will locate 3600 rpm on the N scale and this scale can be finished using a modulus of $m = 1$.

The function of the N scale is $y = 1 \times \log (N/120)$. However, after locating one point (3600) on the N scale, this scale can be completed using the log of N from a log table and neglecting to divide by 120. The factor 1/120 shifts the scale with respect to the other scales. This shift is obtained by locating the value 3600. *Fig.* 4 shows the completed nomograph.

DOUBLE ALIGNMENT CHARTS: A nomograph may be composed of more than three parallel lines. Considering the formula,

$$\frac{A\times B}{C}=D$$

this equation can be transformed into the form: $log\ A + log\ B = log\ C + log\ D$. Since both sides of the equation must be equal (say to value R),

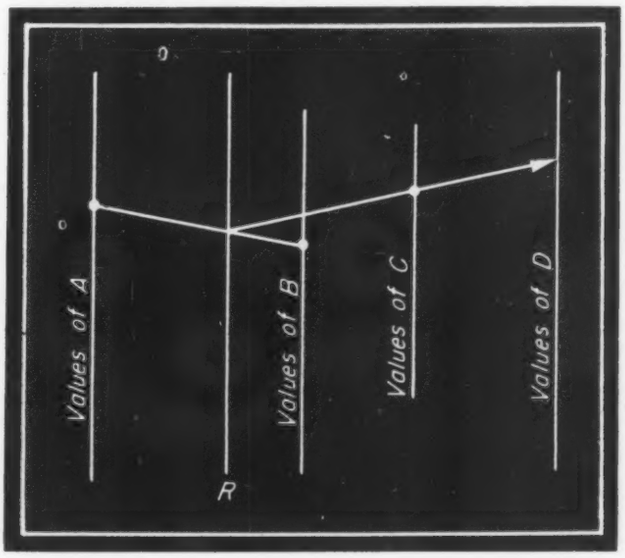
$$log\ A+log\ B=log\ R$$

$$log\ C+log\ D=log\ R$$

Therefore a nomograph for this equation can be designed using the R scale as a "pivot" point. *Fig.* 5 shows such a nomograph schematically. For the construction it is necessary to mark at least one reference point on the R scale. However, in the finished alignment chart the R scale can be entirely without marking.

COMBINED NATURAL AND LOG SCALES: Certain types of formulas require that natural as well as logarithmic

Fig. 5—Schematic chart illustrating use of pivot scale R



scales be used as in *Fig.* 6. The formula for the weight of a steel tube is such a formula. It is

$$W=\frac{\pi}{4}(D^2-d^2)hc$$

where W = weight in pounds, D = outside diameter in inches, d = inside diameter in inches, h = length in inches, and c = specific weight (0.28 lb per cu in.) This formula can be simplified to obtain

$$\frac{W}{0.22h}=(D^2-d^2) \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (A)$$

From Equation A two new equations may be derived, each of which can be used to design a part of the nomograph.

$$D^2-d^2=R \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (B).$$

Fig. 6—Chart for weight of tubing illustrates the transposition from a natural pivot to a logarithmic pivot scale

$$\log \frac{W}{0.22}-\log h=\log R$$

or

$$\log \frac{W}{0.22}=\log R+\log h \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (C)$$

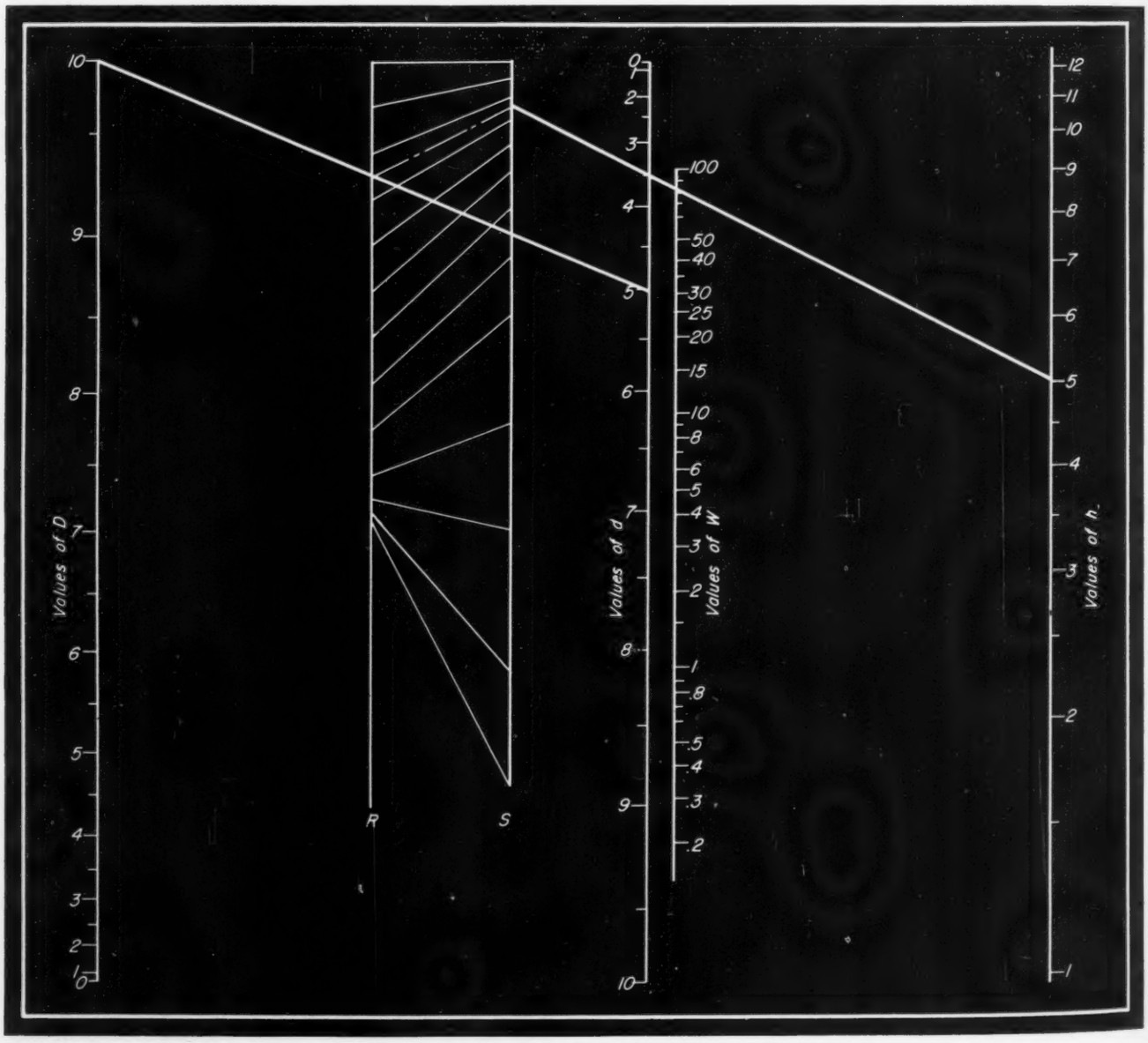
If it is desired that D range from 0 to 10, d from 0 to 10 and both scales do not exceed 5 inches in length, a modulus of 0.05 is obtained and for R scale (m_3) a modulus of 0.025. Two lines are drawn, say, 3 inches apart, obtaining

$$\frac{b}{3}=\frac{0.05}{0.10} \quad ; \quad b=1.5 \text{ inches}$$

The nomograph for equation B can now be designed and, by locating one point on the R scale, this scale can be finished. It should be observed that the d scale is reversed with respect to the D scale as its values are subtracted.

It is obvious that for R in Equation C a new scale is

(*Concluded on Page* 180)



Mechanical Torque—Limiting Devices

By N. C. Weyth
and A. F. Gagne
Development Engineering Division
E. I. du Pont de Nemours & Co.
Pompton Lakes, N. J.

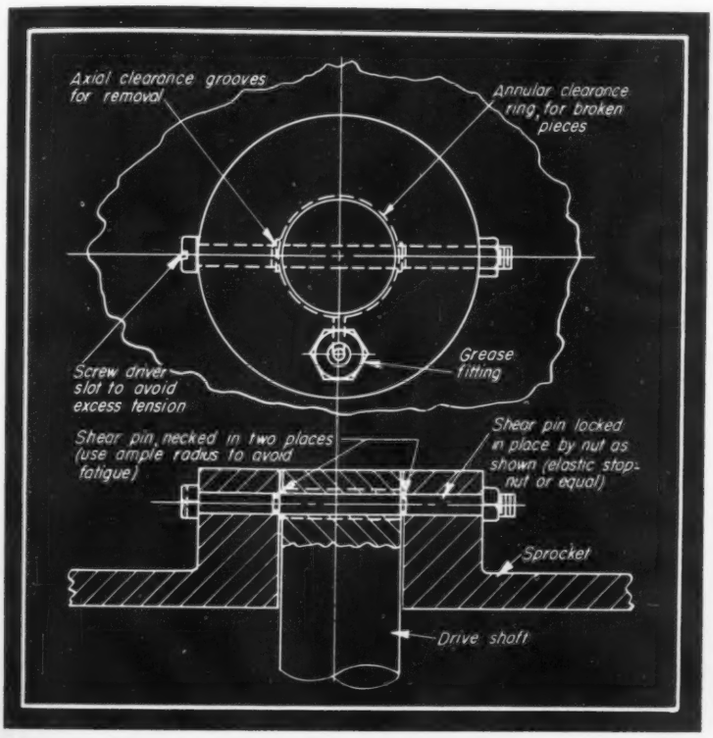

Axial clearance grooves for removal
Annular clearance ring, for broken pieces
Grease fitting
Screw driver slot to avoid excess tension
Shear pin, necked in two places (use ample radius to avoid fatigue)
Shear pin locked in place by nut as shown (elastic stop-nut or equal)
Sprocket
Drive shaft

Fig. 1—Radial type shear-pin coupling. Pin is oversized and necked down to proper section at shear point

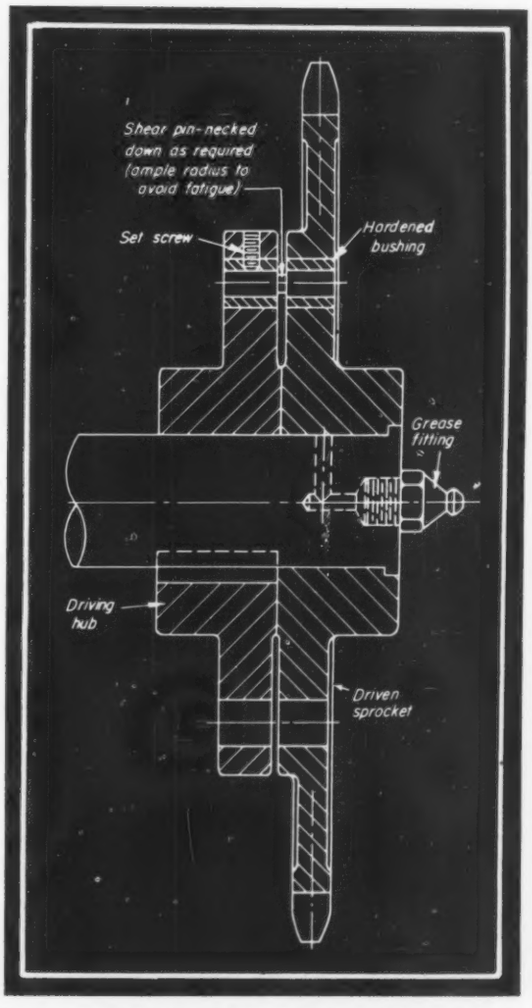


Fig. 2—Axial type shear-pin coupling. Ample clearance between driving and driven members allows pieces to fall clear in case of overload. Lubrication prevents seizing after shearing

WHILE having many advantages, individual motor drives—direct, short-center V-belt or chain—do not provide the effective, if simple, instantaneous overload slip protection and warning inherent in the conventional line-shaft flat-belt installation. The result has been impaired safety and unnecessarily high maintenance costs in many machines. The problem has become increasingly acute in proportion as machines have become more complex and, in tardy recognition thereof, there has appeared on the market a variety of protective devices (mechanical, hydraulic, and electrical) which will transmit no more than a certain preset torque. This article will discuss the more common of the mechanical devices which ordinarily are preferable because of low cost, high dependability, and quick action.

SHEAR-PIN COUPLING: The shear pin is perhaps the simplest, most easily applied, and yet most dependable of any protective device when properly engineered. Like a fuse, however, it

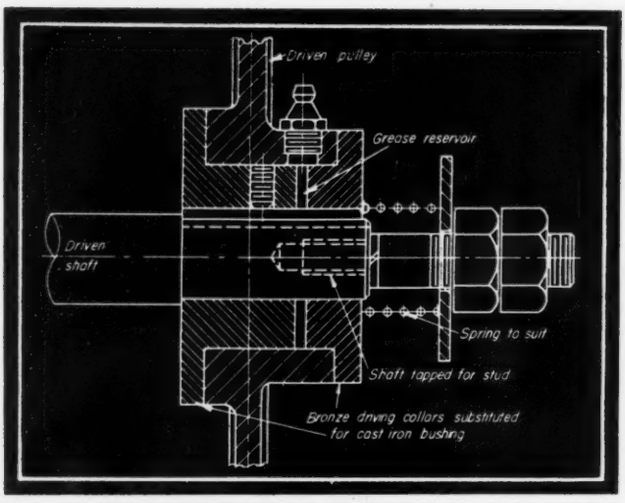


Fig. 3—Above—Adaptation of single-spring slip clutch

must be replaced, and like a fuse there is the same natural tendency to replace with an element of higher capacity with accompanying loss of protection. The shear pin is, therefore, best suited to a drive subject to infrequent overloading only. Selection is not advised in the case of pulsating loads, where the required increase of shear-pin capacity to avoid fatigue failure will reduce sensitivity proportionately. A high-starting torque load will have the same undesirable effect unless eased by a friction clutch or special motor-starting equipment.

Several important design features are illustrated in *Figs.* 1 and 2. In order to give adequate bearing area for the ends and to prevent damage to adjoining parts, the shear pin is always made over-size, with the shear point necked down to suit the load. Shoulders or other special locating means are useful to insure proper location of the necked-down section within the coupling, and an annular clearance ring of generous proportions should be provided between the driving and driven members to relieve the broken

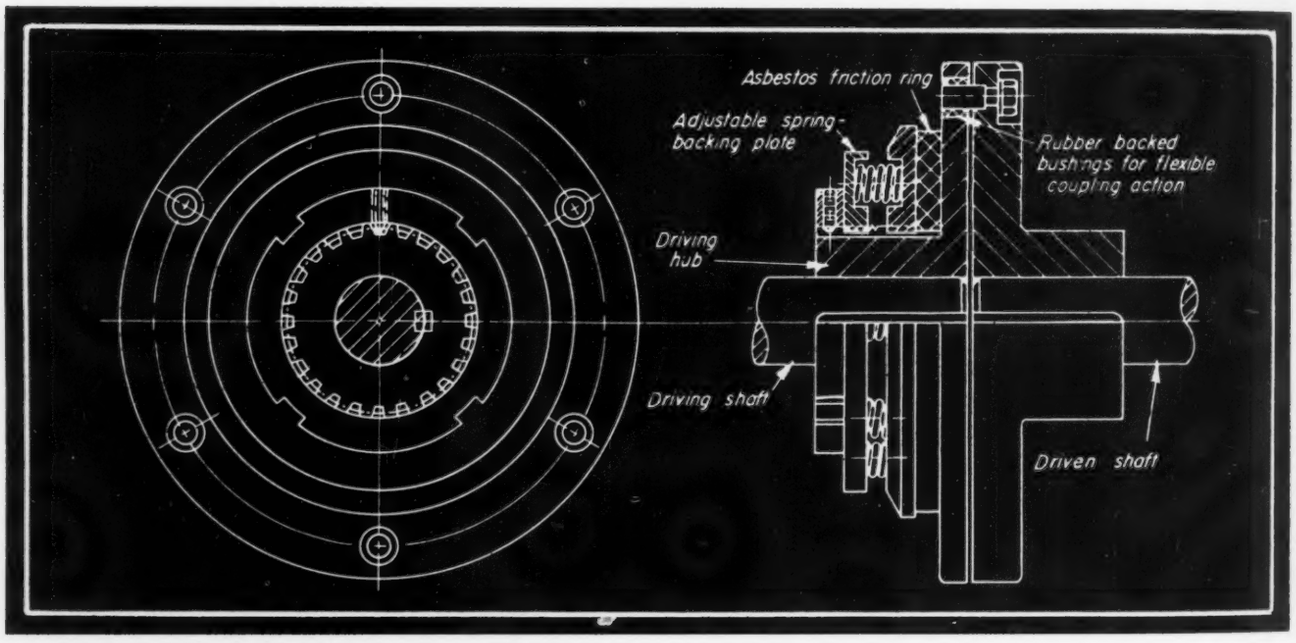
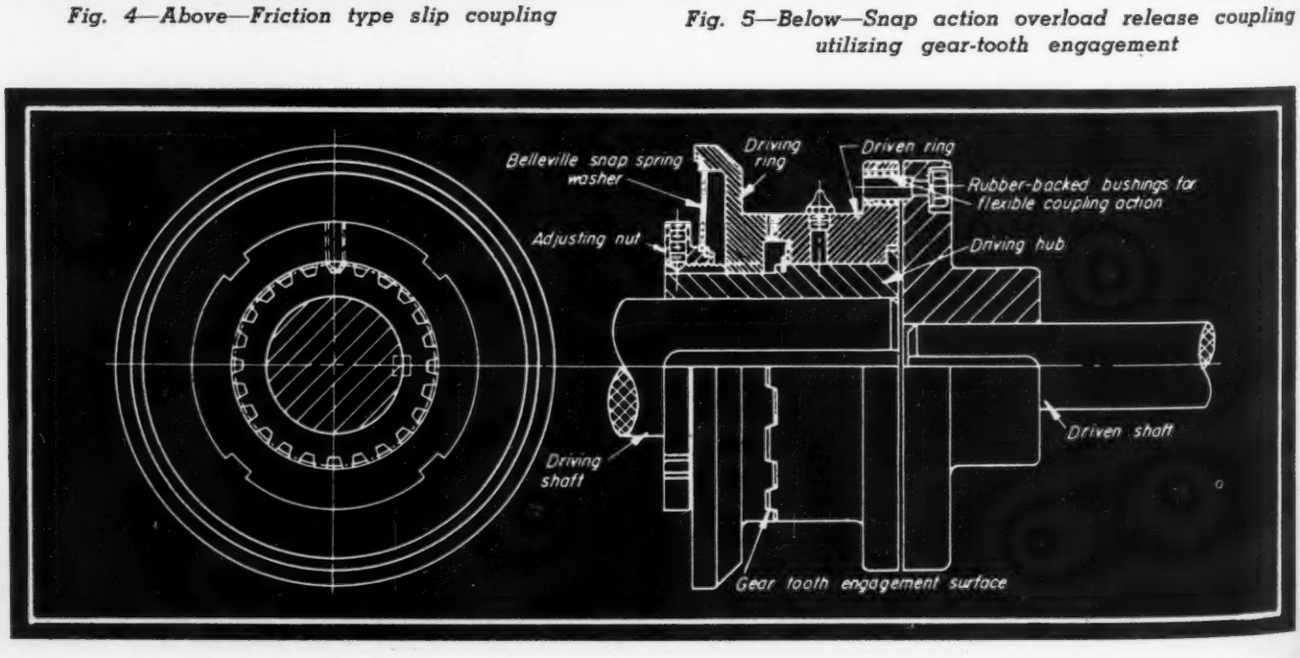


Fig. 4—Above—Friction type slip coupling

Fig. 5—Below—Snap action overload release coupling utilizing gear-tooth engagement



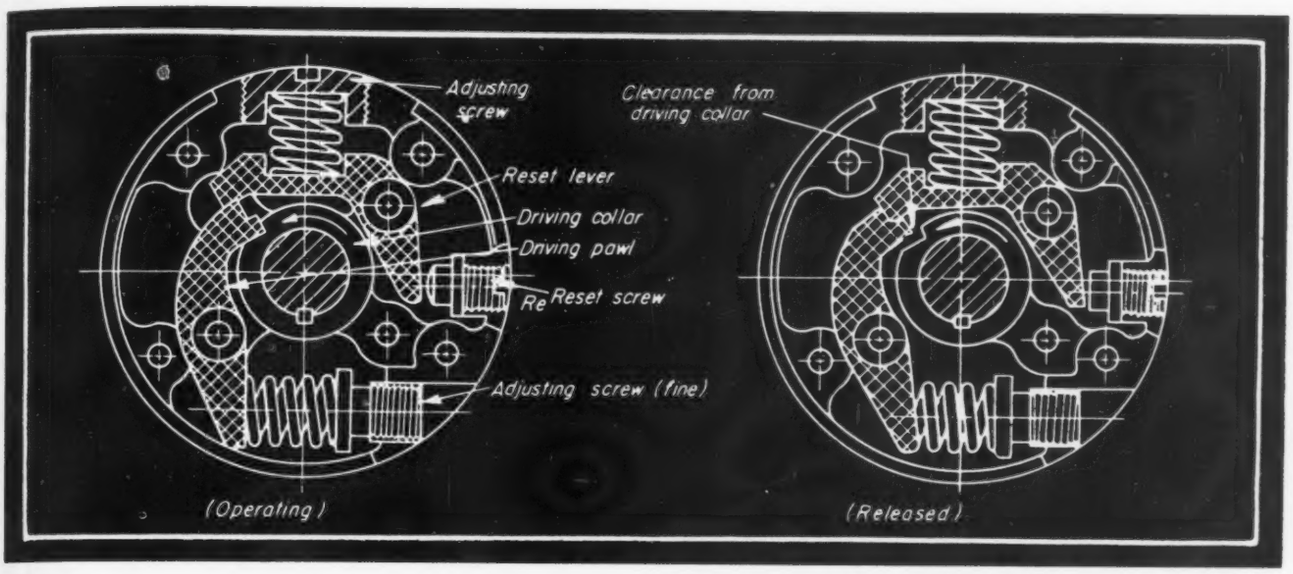


Fig. 6—Snap action overload release clutch employing dog engagement

ends when the pin shears. There should be an allowance for lubrication to prevent galling and seizing after shear, and some provision should be made for controlling the broken shear-pin pieces if these cannot fall clear.

Proper selection of material is important. A hardened, highly-polished steel pin may be chosen to give a sensitive and readily controlled fracture point. For less critical applications, a low-carbon steel (cold-finished SAE-1020) may be selected for its relatively high notch-fatigue resistance. Nonferrous materials are poor because they are too notch sensitive and lack a definite fatigue endurance limit. For highest accuracy, a multiplicity of pins should be used.

Special consideration must be given to proper location of the coupling. It should be accessible for shear-pin replacement, and as with all torque-limiting devices, it should be placed as close to the load as practical in order to minimize inertia effects.

SLIP COUPLING: The slip coupling, in which torque is transmitted through spring or centrifugally loaded friction members, is often a useful alternative to the shear-pin coupling because it does not require replacement at every overload, because it is not affected by load pulsation but acts instead to damp shocks, and because it can be adjusted readily. The characteristic controlled slippage is of special advantage when it is desired to pick up the load

Quick Selector Chart—Mechanical Torque-Limiting Devices

Type of Coupling	Manufacturers of Typical Units	Relative Cost	Dependability	Necessary Load Characteristics	Reset	Limitations	Special Advantages
Shear-pin Coupling	(Available from most manufacturers)	Low	High	Light to moderate starting torque Non-pulsating load	Replacement required	Must be accessible for replacement Undesirable in case of frequent overload Not readily adjusted	Easily applied
Slip Couplings	Hilliard Falk Flaton	Moderately high	High	No limitations	No release of set torque	Continues driving after overload May be dangerous, will overheat Frictional torque not fixed—may vary	Automatic cyclic adjustment of torque possible Gives controlled acceleration start-up
	Rawson Newaygo Engineering Co.	Moderate	Very high	Above 600 rpm	No release of set torque	Continues driving after overload, will overheat Not readily adjusted Torque affected by speed	Gives controlled acceleration start-up
	Bethlehem	Moderate	Very high	Above 375 rpm	No release of set torque	Continues driving after overload, will overheat Not readily adjusted Torque affected by speed	Gives controlled acceleration start-up Precise torque control at given speed
Spring-Loaded, Detent Couplings	Hilliard	Moderately high	Very high	Light to moderate starting torque	Manual reset	Must be accessible for reset	Safety snap-out action Very low maintenance Easily adjusted
	Overload Release Clutch Co.	Moderate	High	Light to moderate starting torque	Manual reset	Must be accessible for reset	Safety snap-out action Easily adjusted Locks in one definite angular relationship
	A. Y. Dodge Co.	Moderately high	Fair	Light to moderate starting torque	Automatic reset when driver is stopped	Running speed must be above 500 rpm Not readily adjusted Torque affected by speed	Safety snap-out action Sensitive to overspeed as well as torque (Proportion adjustable between 10% and 90%)

slowly despite a fast-starting motor, or when a cycle variation in torque is desired, as in wind-up equipment. Continued application of torque after release sometimes may be dangerous to equipment or personnel, in which case a standard plugging switch can be used to sound a warning or shut off the drive.

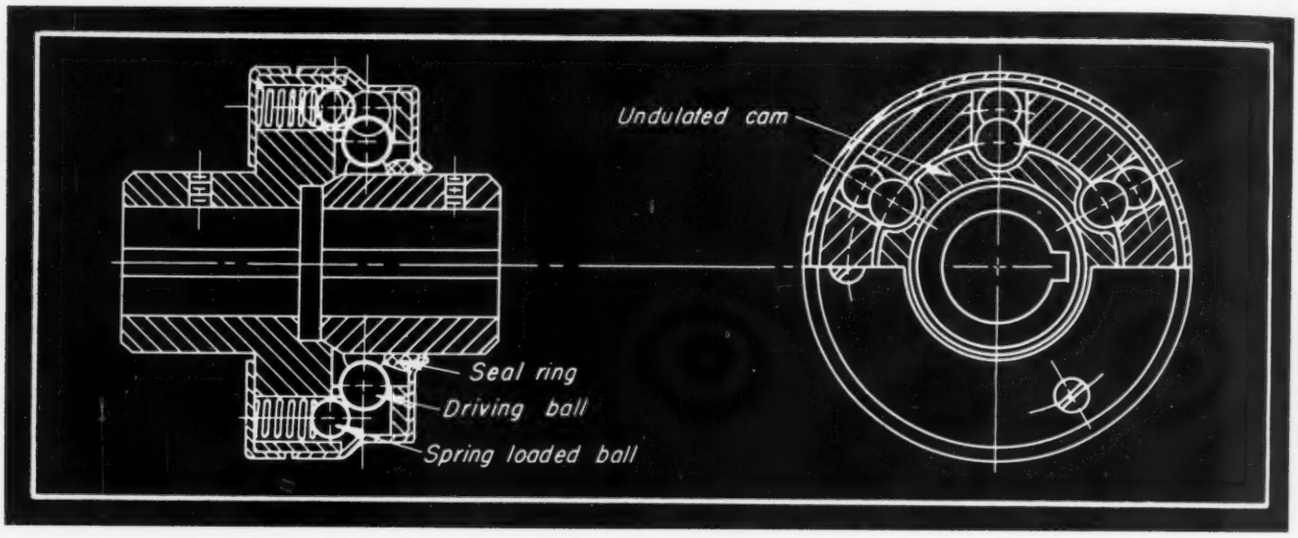
Construction used successfully to build the action into an existing light V-belt drive is illustrated in *Fig.* 3. The unit in this case is termed commercially a "clutch" because it carries a sheave; the "coupling" connects two shafts.

Most commercial couplings are provided with flexible coupling action in addition to the slip action as illustrated in *Fig.* 4.

OVERLOAD-RELEASE COUPLING: There are available a variety of commercial over-load-release couplings based on mechanical detents supported by springs. In order to prevent hammering of the mechanism after release, usually a snap action requiring manual reset is incorporated. This is often a valuable safety feature, although it may cause a certain amount of operating annoyance. The positive drive is useful to prevent lost production through slip and is invaluable for timing purposes; however, the coupling should not be used where a high-inertia load is connected to a line-start motor because, like the shear-pin coupling, sensitivity will be impaired when the springs are adjusted to carry the lurch of starting.

Several different commercial couplings are illustrated in *Figs.* 5, 6 and 7. The Hilliard coupling, *Fig.* 5, uses a Belleville-spring washer in a design admirable for its simplicity and sturdiness. The coupling is readily adjusted. The Overload Release Clutch Co. device shown in *Fig.* 6 is also sturdily made, easily adjusted and locks in one definite angular relationship for timing. Modified characteristics are found in the A. Y. Dodge Co. ball-detent coupling, *Fig.* 7, which may be made from 10 to 90 per cent sensitive to overspeeding. Reset after release is automatic, but requires stopping of the prime mover in keeping with best practice. This coupling is limited to speeds over 500 revolutions per minute.

Fig. 7—Centrifugal action overload release coupling with ball detent engagement



The accompanying quick-selector chart tabulates the principal characteristics of shear-pin, slip and overload-release couplings.

The authors are indebted to Mr. William Staniar of the Du Pont Co. for his helpful suggestions in the preparation of this paper.

ASME Machine Design Group To Meet in Detroit

First technical session of the newly formed Machine Design group of the American Society of Mechanical Engineers will be held on the afternoon of Monday, June 17, 1946, during the semiannual meeting of the Society in the Hotel Statler, Detroit. All engineers interested—whether A.S.M.E. members or not—are cordially invited to attend. A nominal registration fee is required of nonmembers. Following is the program for the session:

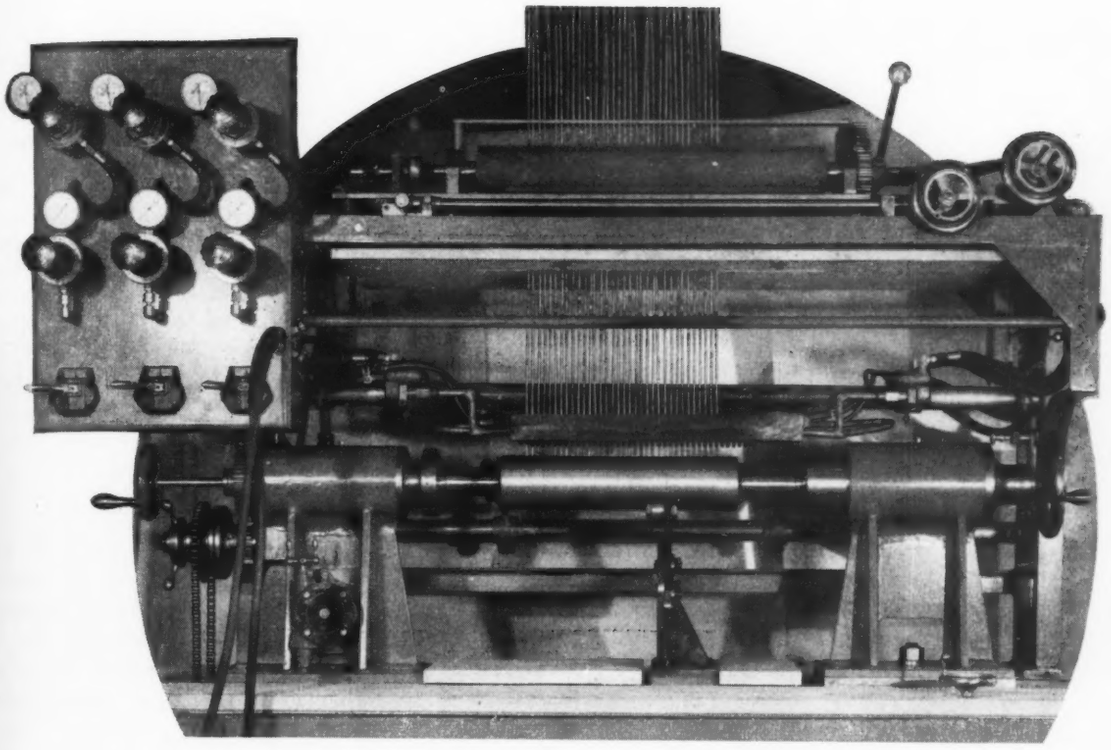
"Application of Tables for Helical Spring Design"
H. F. Ross, United Shoe Machinery Corp., Beverly, Mass.

"High-Speed Chain Developments"
A. W. Meyer, Brown & Sharpe Mfg. Co., Providence, R. I.

"Evaluating Self-Locking Nuts for Industrial Equipment"
Wm. E. Horenburger, Elastic Stop Nut Co. of America, Union, N. J.

Chairman at the session will be H. L. Keller, Buick Motor Div., General Motors Corp., Flint, Mich.

Started in 1945, the ASME Machine Design group provides a common meeting ground for machine designers in all branches of industry and, by sponsoring papers such as the foregoing, aids in the dissemination of technical information of special interest to designers. The group this year is under the chairmanship of J. F. Downie Smith, head of the engineering department at the Research Div., United Shoe Machinery Corp., the secretary being B. P. Graves, director of design, Brown Sharpe Mfg. Co.



Localized Surfacing Combats Wear, Corrosion

By A. R. Lytle

Union Carbide & Carbon Research Labs., Inc.
Niagara Falls, N. Y.

Fig. 1—Above—Multiple rods fed by gravity during the weld-cycle, deposit a uniform hard surface over the entire length of cylindrical parts

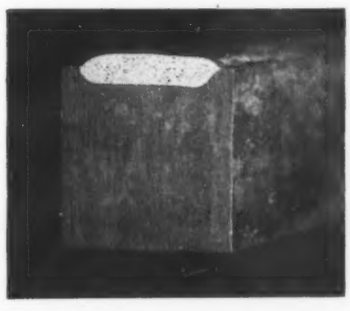


Fig. 2—Section through a guide bar showing hard surfacing metal deposited within a groove machined in the top surface

LOCALIZED surfacing of machine parts for increased resistance to corrosion and wear, and many times for other special purposes, is now not only accepted but often demanded. In fact, many parts are designed so that specific physical or chemical properties may be given to localized surfaces through the proper application of surfacing metals or alloys. Typical examples of such tailored-to-use assemblies are valves and valve seats of all sorts, bearing faces, bushings, sleeves, dredger teeth, sprocket teeth, drill bits and the like.

Localized surfacing can be accomplished by metal spraying, by applying preformed pieces using mechanical fastenings, by brazing or low-temperature soldering, and by fusion welding, *Fig.* 1. However, fusion welding is by far the most versatile and widely applicable of these methods since it is not limited by size, shape thickness, machinability, or numerous other factors that may limit other methods. Moreover, fusion welding permits a choice of the full range of metal or alloy surfacing materials and unites them in a perfect welded relationship to the underlying base metals.

Fusion welding can be further divided into electric arc and flame welding.

From a paper presented at the twenty-sixth annual meeting of the American Welding Society.

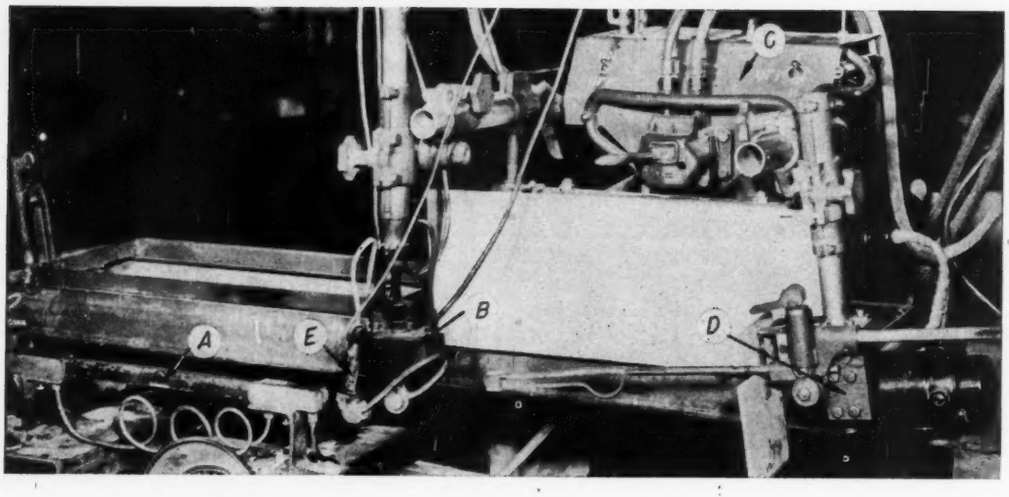


Fig. 3 — Hard surfacing machine for oil expeller barrel bars. Prebent bars A are held in water-cooled supports, heating head B is mounted on motorized carriage C, with motorized rod feed at D and air jets for controlling contour of deposit at E

but most attention has been given to the flame-welding process, because it has many special advantages for surfacing and as a consequence is widely used. Some of the most important technical characteristics of this process as applied to surfacing operations are as follows:

1. Minimum dilution of deposit with base metal
2. Carbon can be added to carbon hardenable alloys
3. Surfacing metal can be sweated to base metal without deep melting
4. Process is equally useful for applying most surfacing alloys including brass, bronze, Haynes Stellite rod, high-chromium alloys, and cast hard carbides
5. Deposited metal can be smoothed by flame manipulation, reducing finishing costs
6. Thickness of deposit can be less than 0.030-inch
7. Residual heat retained in base metal permits smoothing deposit by hot working
8. Rate of cooling after welding does not harden base metal
9. Completed surface is continuously visible during welding
10. Process can be mechanized.

It is not the purpose of this article to enter into a detailed discussion of the first nine items except to emphasize briefly the relationship that exists between the characteristics of the flame-welding process and its widespread use for surfacing operations. The advantages of mechanization, however, are manifold and merit discussion at this time.

Mechanized processing is likely to show greatly reduced cost for many reasons: It requires fewer and less well-trained operators; the product is much more uniform and can be made to conform closely to requirements; and there are fewer rejects. Mechanization of the surfacing operation is the finishing touch whereby almost perfect localized surfacing can be accomplished repetitively on a large scale.

There is no hard or fast rule to determine how many parts per day will justify mechanization because much depends upon the size, ease of handling, distribution of the hard-surfaced areas, and type of deposit. Nevertheless, in most cases a progressive, imaginative engineer can determine the feasibility and broad economics of making the process automatic.

Much can be done in placement or control of the disposition of the surfacing metal by judicious arrangement of the several welding flames, but generally for mechanized operations definite limiting barriers are of much more practical value. These may take the form of shallow or deep grooves of a depth equivalent to the depth desired in the deposited metal. These have been found to be quite satisfactory for positioning the weld metal within close tolerances, *Fig.* 2. There are, of course, instances in which the surfacing material is to be applied to an exterior corner of a piece where a groove is not possible. In such cases, a refractory plug or mold

Fig. 4—Left Below—Transverse section through bronze-faced cast iron guide bars before machining. Shown 1.5 times normal size

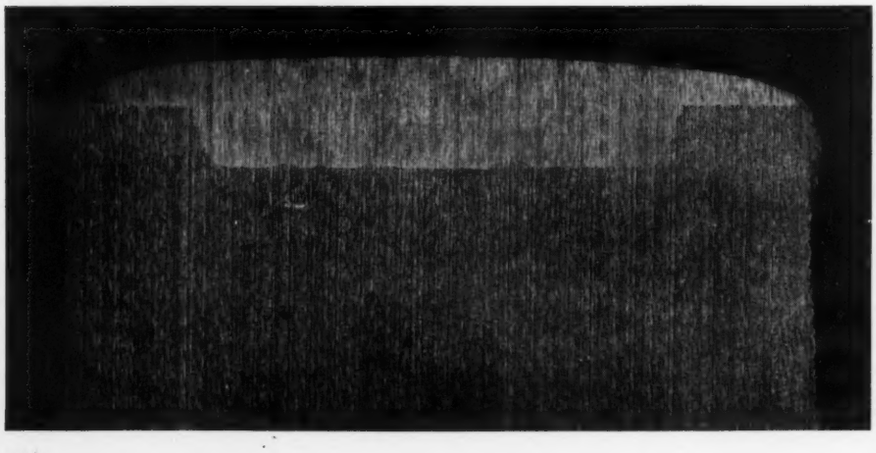
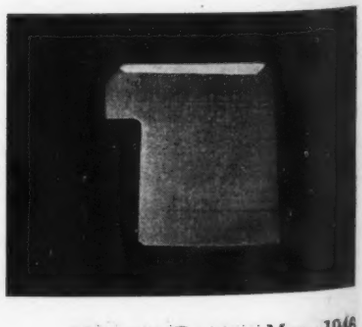


Fig. 5—Below—Section through a seat ring after machining to size



is entirely satisfactory even for some relatively large-scale surfacing operations.

The several methods that have been developed for surfacing operations are referred to as the straight-line, cylindrical and spiral types with, of course, some variations and combinations for particular installations. Hard facing of oil expeller barrel bars is a good illustration of the straight-line method. Bars are ½-inch wide, 1-inch thick and 11 inches long, and a deposit of Stellite alloy is required on the ½ x 11-inch face. Hundreds being required per day, mechanization was well justified. As welding conditions can be set up for almost any predetermined rate of production, a reasonable rate was chosen, in this case 5 inches per minute. A multiple-flame head was fixed with flames so directed to bring the base metal to proper sweating temperatures and melt the rod, distribute the molten metal to the desired points, using a mechanical rod feed with accompanying rod preheat set up as shown in *Fig.* 3.

Two difficulties developed during this work. First, because of the high surface tension of the molten metal, the deposit tended to hang over the edges, whereas a square-faced deposit was desired in order to minimize grinding. By directing a fine stream of air against the side of the bar just below the molten metal this slight overhang was eliminated and only a minimum amount of grinding was required for finishing. The second problem was warpage. As might be expected, the application of a layer of molten metal to one surface of a relatively small bar set up differential cooling stresses, with the result that the bars warped, the deposited metal being on the concave side. Therefore, the bars were prebent in the opposite direction by an amount equivalent to that which developed during the surfacing operation and as a result of this prebending, the surfaced bars were straight as they came out of the machine.

This method has also been used for applying a surface of bronze to cast-iron bars for such parts as guides. Bars 24 inches long by 2 inches square, with a 1⅜-inch wide by 3/16-inch deep deposit of hard bronze, were desired. For this operation the same general principles were applied as outlined previously except, of course, that no prebending or preforming was needed. The weld metal was provided by ⅜-inch diameter bronze welding rods fed-in mechanically and fluxing was accomplished by a combination of vapor flux and a flux coating on the rod. This combined fluxing gives excellent results for bronze-welding operations, *Fig.* 4.

The general principles of straight-line surfacing can also be applied to the surfacing of other products when a relatively long, narrow surfacing operation is required. A typical operation is the surfacing of the seat rings for large-diameter gate valves, *Fig.* 5. In *Fig.* 6 is shown a production machine for the facing of rings in this manner. This machine has been satisfactorily used for applying both Stellite and hard bronze to the seating surface of rings that range in diameter from 6 to 18 inches, with deposits from ⅛-inch deep by ½-inch wide to ¼-inch deep by 2 inches wide.

Widespread demand for hard-faced parts such as pump rods and sleeves, bushings, collars, and crushing rolls for

Fig. 6—Above—Machine for hard facing large annular parts automatically with capacities ranging from 2 to 18-inch diameters

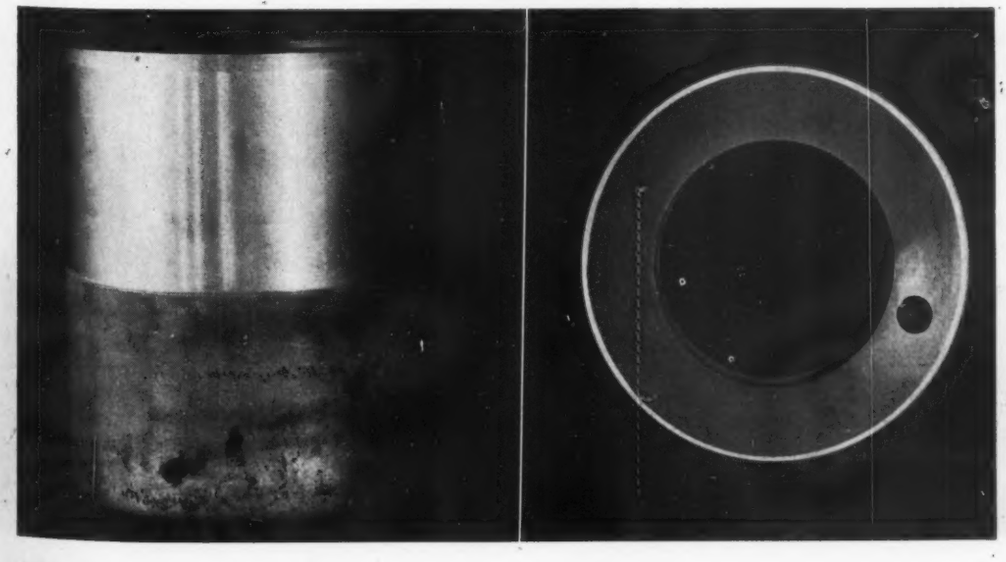


Fig. 7 — Left — Specimen showing smoothness of cylindrical welded deposit, soundness after machining and uniformity of thickness

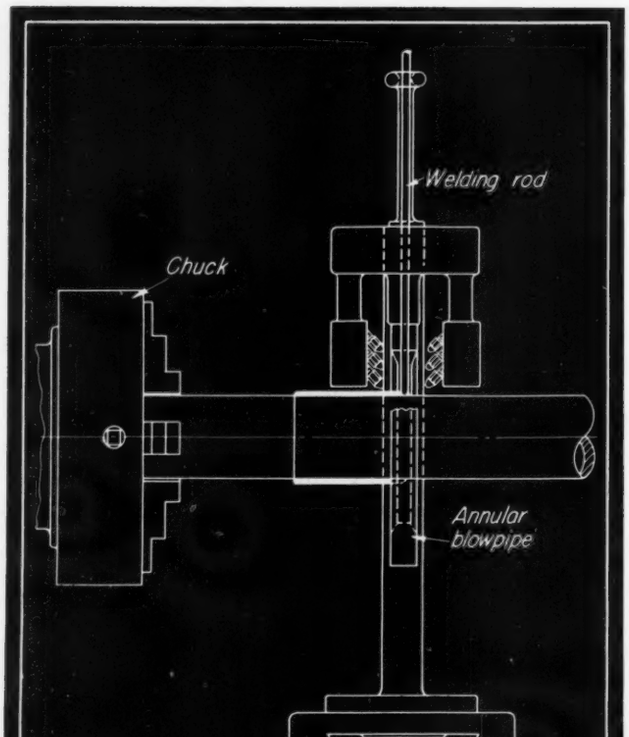


Fig. 8—Schematic layout showing spiral method used for the surfacing of long or thin-wall cylinders

the oil and food processing industries, has brought special attention to the development of suitable methods for surfacing this type of work. Thus far two satisfactory methods have been developed; cylindrical and spiral types.

The principal characteristic of the cylindrical method is that the entire length of a cylindrical area to be covered is coated in one revolution of the work. This is simply an adaptation of the straight-line method to a cylindrical surface in which the cylinder to be coated is supported horizontally and rotated slowly while molten weld metal is deposited simultaneously over the entire length of the work. Details of the apparatus and technique to accomplish this method of hard facing are shown in the head illustration, *Fig.* 1.

For this surfacing operation, a series of preheating flames as long as the section to be surfaced is placed 90 degrees ahead of the line of welding. To apply the entire coating in one revolution and distribute it uniformly, multiple rods are required. Each rod passes freely downward by gravity through a tube and between two converging flames that impinge against the rod just above the point of contact with the work. Also, the entire table supporting the rotating spindle may be oscillated lengthwise. This slight oscillation is desirable to overcome the furrows that develop on the surface between the individual rods when no oscillation is used. The stroke is equivalent to the spacing of the rods and serves to level off the molten metal before is solidifies. The deposit as it leaves this station is smooth, of uniform depth and sound, *Fig.* 7.

The cylindrical machine discussed has been used for such parts as heavy-wall sleeves and bushings up to 4½ inches in thickness and 11 inches in length. Other machines of the same type have been used for hard surfacing sleeves as small as ⅞-inch diameter by 3 inches long and for heavy collars 8 inches in diameter by 6 inches long, having a 1-inch wall. Moreover, by changing the gas pressures on the several heating heads and adjusting the position of the head in respect to the top center, high-quality deposits have been made with a variety of surfacing alloys ranging from the conventional hard bronze or brass to Stellite alloys and various high-chromium alloys. The cylindrical method is well suited to the surfacing of solids and heavy-walled cylinders up to about 11 inches in length, but some difficulty is encountered in maintaining proper heat balance when the process is extended to the surfacing of longer or thin-wall cylinders. For this latter type of work, the spiral method has been developed.

Basically the spiral method is another modification of the straight-line method, but in this case the deposit is applied as a narrow ribbon in a continuous fine-pitch

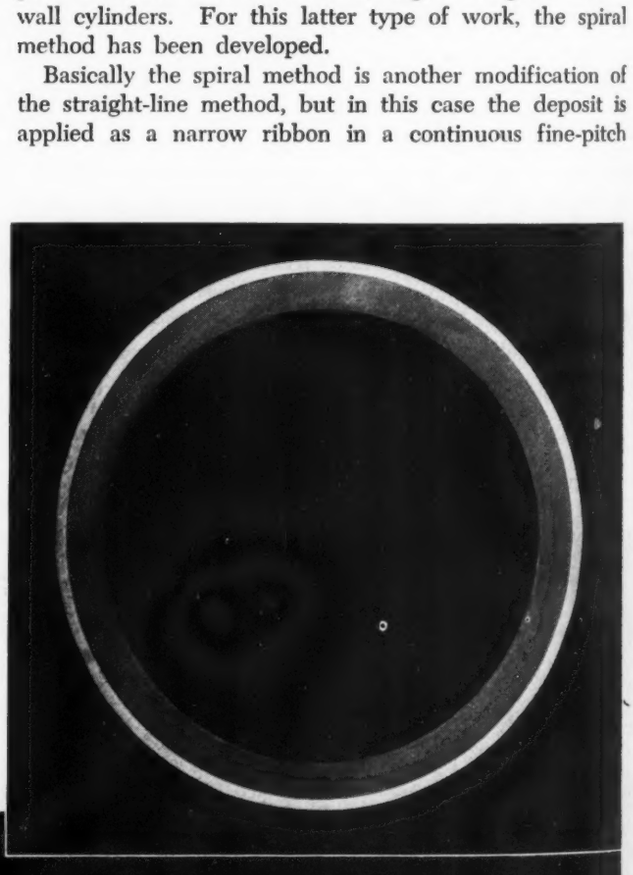


Fig. 9—Below—Sections through thin-wall tube showing uniformity of section and absence of warping

spiral so that each successive spiral overlaps the previous deposit and forms a continuous surfacing layer. There are, of course, many ways in which this might be done, but most success has been met with the setup which is shown in *Fig.* 8. Preheating and welding heads and rod guide are mounted on a separate traversing carriage moving parallel with the axis of the work. The preheating head is annular in shape, surrounding the work and providing a series of inwardly directed flames. These flames not only preheat the base metal but also, because of their reducing nature, protect it from oxidation.

Tests have been made at surface speeds as high as 1000 surface feet per minute without any observed tendency for ejection of the metal, and at surface speeds as low as 250 surface feet per minute without having the molten metal accumulate at the bottom. Another advantage of the high rate of rotation is that the advance or pitch of the spiral is on the order of 0.010-inch. With this fine pitch there is practically no furrowing or spiraling on the surface of the deposit, and the metal solidifies continuously as the welding and preheating heads advance. Solidification takes place about ⅜-inch axially behind the point at which the welding rod is added, and by the time the metal reaches this point the weld metal has had ample opportunity to free itself from dissolved gases and hence solidifies in a sound, smooth deposit with a high degree of uniformity of thickness around the periphery.

High Rate of Deposition

Depth of the surfacing layer is determined by a combination of rate of melting and rate of traverse of the rod lengthwise. Variation in traversing speed is the most convenient means of controlling the depth of deposit. Experience has shown that for a cylinder 3¼ inches in diameter, a depth of 0.100-inch can be readily and consistently applied at a traversing speed of ¾-inch per minute. At this speed, welding rod is melted at the rate of about 16 pounds per hour, which is a high rate of deposition for weld metal. Such a deposit cleans up on machining to 0.080-inch, which is as thick as necessary for many hard-surfacing operations when corrosion or relatively moderate wear is to be overcome.

One of the important characteristics of this method of surfacing is the marked freedom from warping of thin-wall cylinders, *Fig.* 9. These are sections taken through a 3¼-inch diameter, ¼-inch wall cylinder 11½ inches long. This cylinder, after normal air cooling, still ran true. Caliper measurements of the bore indicated a radial shrinkage of 0.020-inch, which was constant for the length of the deposit, but otherwise no dimensional change had occurred.

Continuous surfaces represent the bulk of parts on which surfacing is required for corrosion resistance, bearing faces and the like. However, a large amount of hard facing is done on intermittent surfaces, but to date no satisfactory fully mechanized method has been developed for this type of operation. The difficulties of such a development are obvious in light of the principles outlined previously, but good progress has been made in placing this type of hard surfacing on a semiautomatic basis.

Most intermittent surfacing is associated with toothed parts such as drill bits, coal mining undercutter bits, etc. For such surfacing tungsten carbide alloys are frequently chosen. Experience and practice in the past have been divided between the use of large inserts of carbide embedded in steel weld metal and the use of fusion-welded surfaces containing small grains of the carbide. In many cases of intermittent application a large solid insert is not needed, it being necessary to deposit only a very thin layer of the hard tungsten carbide. Therefore, a new method has been developed in which crushed granules of cast tungsten carbide are firmly welded superficially onto a surface that has been heated to sweating temperature by the excess-acetylene welding flame. This method results in the formation of a thin layer on which

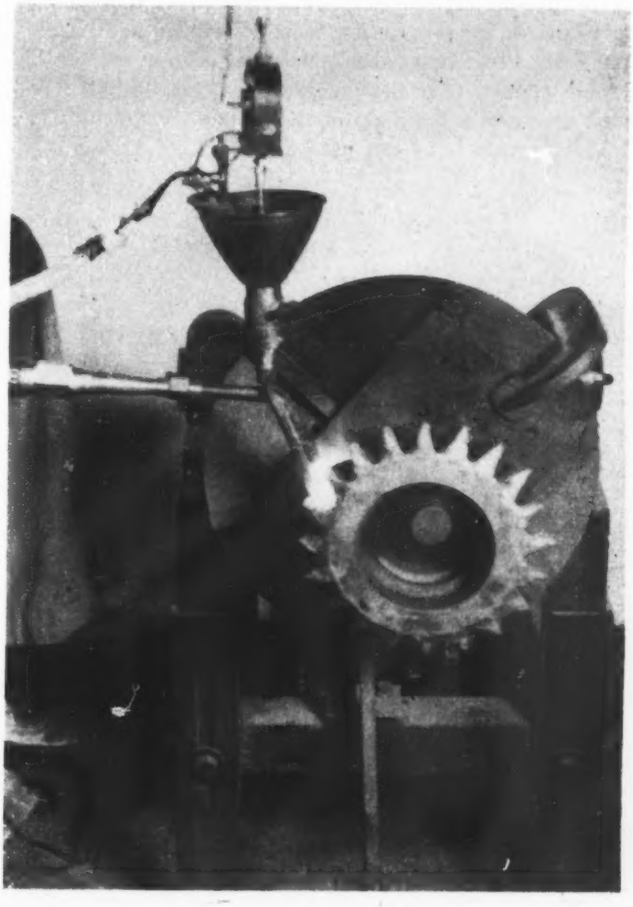


Fig. 10—Manually operated apparatus used in applying a wear resistant surface of tungsten carbide granules

the granules may either be submerged or actually allowed to protrude above the surface. The conditions may be readily adjusted so that the deposit can be applied to thin or sharp edges or points without materially altering their contour or sharpness.

Apparatus used for this type of surfacing is illustrated in *Fig.* 10. It consists of a blowpipe supplying mixed oxygen and acetylene to a standard single flame or special welding tip and a grain dispenser connected by flexible tubing or directly attached to the welding head. Central passage in the tip is connected directly with the grain dispenser, which with the aid of a vibrator provides a controllable and uniform flow of granules.

Operation of the apparatus is relatively simple. On

small parts such as coal bits, the area to be surfaced is brought rapidly to a sweating temperature and then the flow of the tungsten carbide granules is started. In a few seconds a sufficient depth of surfacing material is built up and the operation may be stopped. If the workpiece is larger, the operation is started as described but the blowpipe is advanced in a slight weaving motion, building up the deposit to the desired height in one pass. In certain instances of heavy work some preheat is beneficial as it permits more rapid advance and smoother deposits.

As may be expected, the operator has considerable latitude in respect to the depth of deposit that may be built up and in the proportion of base metal melted and mixed with the granules of tungsten carbide. By rapid advance of the blowpipe, a superficial deposit can be made, or by progressing more slowly or with higher gas flow, some base metal may be melted to form a smooth coating over the granules and thus embed them more securely in the body of the metal.

This process has been used for the hard facing of a

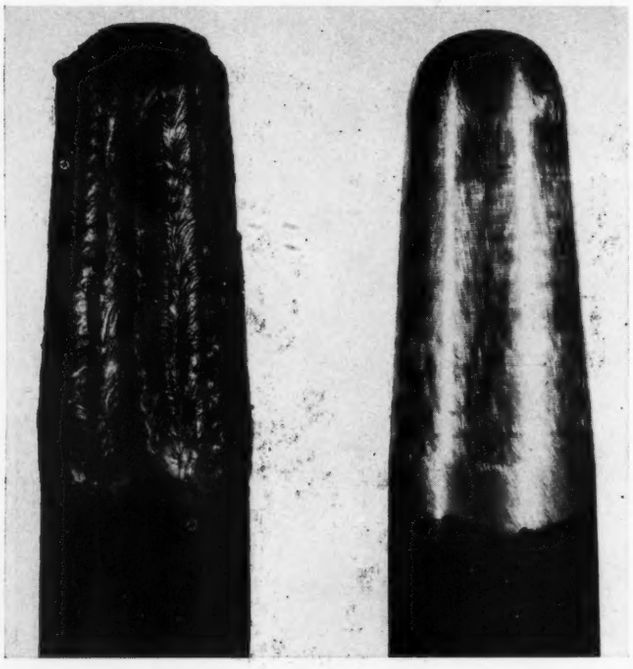


Fig. 11—Hot piercing punch for 75-mm shells faced with Hastelloy C increased production 400 per cent over that obtained with tool steel

widely diversified group of parts exposed to severe wear, and satisfactory performance has been obtained. Typical examples where the granules are applied superficially and thus are in the most favorable position to resist abrasion, are earth scraper blades, scarifier teeth and star rock bits. In other applications such as plowshares, hay-cutter blades, tank treads, sprocket teeth, etc., it may be desirable to embed the granules so as to have a smooth finished surface. In such surfacing deposits, the granules of tungsten carbide are not visible.

Additional advantages of the process are economy in the use of the granules and speed of application. For work such as scraper blades or hay-cutter blades, the surface may be coated at the rate of 3½ square inches per minute using fine-mesh (60 by 100) granules at the rate of 0.005 pounds per square inch of area coated. Coarse grains (24 by 36) are used at the rate of 0.01 pounds per square inch.

No discussion of modern hard-facing methods would be complete without a description of the use and value of the high nickel-base Hastelloy alloys. These alloys are well known for their special corrosion resistance and retention of high strength at elevated temperatures, and they are used extensively throughout industry for especially severe corrosive conditions. Recently, however, Hastelloy alloy C has demonstrated remarkable qualities of resistance to abrasion and spalling at high temperature and as consequence it has found an additional field of service as a hard-facing material.

Hastelloy alloy C can be applied satisfactorily by electric arc welding using either a coated cast rod or a composite rod in which some of the alloying ingredients are included in the coating, or by the usual flame-welding technique. As deposited by either process it is relatively soft, having a Rockwell C hardness of about 18. The deposited metal is, therefore, readily machined by conventional methods and does not require grinding as do hard-alloy deposits. However, the metal does not soften or lose its properties at elevated temperatures. The alloy also has a high resistance to seizing or pin-point welding at both low and high temperatures and hence is relatively free from the fretting and galling types of wear characteristic of many other soft metals used for surfacing. Since the deposited metal is relatively soft, it has been found by experience that the residual thickness of deposit after machining usually should not be greater than 1/16-inch. This thickness of material gives ample protection and has the further advantage that little base metal has to be machined away before surfacing.

Tremendous Increase in Production

Typical examples of especially satisfactory applications have been shell punches and drawing dies, draw rings for seamless boiler tubes, roller twist guides for hot rolling of bar stock, and hot shear blades. In one particular case an 8-inch diameter hot-pierce punch 24 inches long was coated with 1/16-inch of Hastelloy C for a distance of 9 inches from the nose, *Fig.* 11. After this had been machined to finish size, a 100 per cent increase in production was obtained as compared with the former deposit of the hard alloy type. In another typical installation of a hot-shear blade, the average life of the blade was increased from two days to five weeks.

Another recent development is the application of these nickel base alloys to parts subjected primarily to corrosion, where a surface layer of one of the corrosion resistant Hastelloy alloys on a steel base metal is sufficient to withstand the corrosive action. Many parts have been given greatly increased life by this procedure. Shafts and shaft sleeves for pumps handling hydrochloric and sulphuric acids are typical. Hastelloy alloy B gives excellent service in contact with hydrochloric acid and satisfactory coating may readily be obtained by metallic arc welding with a coated electrode. For sulphuric acid, Hastelloy alloy D applied by flame welding is recommended. In either case, the coating after machining need be only about ⅛-inch thick.

Fig. 1—Test method of clamping aluminum springs in compression. Shown from top to bottom: Completely compressed with fiber stress of 50,000 psi, partially compressed at 25,000 psi, and free length

An Aluminum Spring May Be The Answer!

By Amiel Gelb
Chief Testing Engineer
Reynolds Metals Co.
Louisville, Ky.

ALTHOUGH aluminum has seldom been considered suitable for spring applications, the advent of war-developed high-strength aluminum alloys warrants a careful re-analysis. The evaluation of the suitability of one of the new aluminum alloys, attempted in this article, indicates that this alloy possesses eminently suitable characteristics for effective utilization in springs such as shown in *Fig.* 1. Experimental data are presented to aid the designer in determining the applicability of a new aluminum alloy for employment in springs.

The zinc-magnesium type aluminum alloy, known as R303, is of particular interest* in this connection because of its exceptionally high resilience, having a typical yield strength in excess of 70,000 psi. The metallurgical characteristics and mechanical properties have recently been comprehensively described† and, therefore, only a minimum of repetition is required. In common with the family of commercial wrought aluminum alloys, R303 is characterized by a low specific gravity (2.82), so that this material weighs approximately one-third as much as equal-volume sections of the more common structural metals, such as steel, brass, or copper.

Optimum mechanical properties are developed in this material by heat-treatment at 820 F, followed by quenching in water and artificial aging at either of two temperatures. When aged for 8 hours at 315 F, the alloy is designated as being in the T315 temper, and when treated at 275 F for 25 hours its designation is T275 temper. The

* The aluminum alloy 75S is known to have similar characteristics. This discussion is confined, however, to R303, since the experimental investigation was limited to that alloy.

† "Aluminum-Alloy R303"—P. P. Zeigler, L. E. Householder and H. N. Logsdon, *Iron Age*, 74, October 4, 1945.

TABLE I—Typical and Guaranteed Minimum Tensile Properties of R303 Alloy

	Typical			Tentative Guaranteed Minimum		
	Tensile Strength (psi)	Yield Strength 0.2% Set (psi)	Elongation (% in 2-in.)	Tensile Strength (psi)	Yield Strength (psi)	Elongation (% in 2-in.)
Sheet*						
Annealed (0 temper)	30,000	15,000	18	35,000†	—	10
Heat-treated and aged						
T315 temper	75,000	69,000	9	70,000	65,000	7
T275 temper	79,000	73,000	9	74,000	69,000	7
Extrusions						
Annealed	30,000	15,000	18	35,000†	—	12
0.040-in. to 0.600-in. thick						
T315 temper	79,000	73,000	9	72,000	67,000	7
T275 temper	85,000	80,000	9	77,000	72,000	7
0.601-in. thick and heavier						
T315 temper	82,000	77,000	11	75,000	70,000	8
T275 temper	89,000	84,000	10	80,000	75,000	8
Rolled Rod up to 4-in.						
Annealed	30,000	15,000	24	35,000†	—	16
T315 temper	75,000	69,000	14	70,000	65,000	10
T275 temper	79,000	73,000	14	74,000	68,000	10

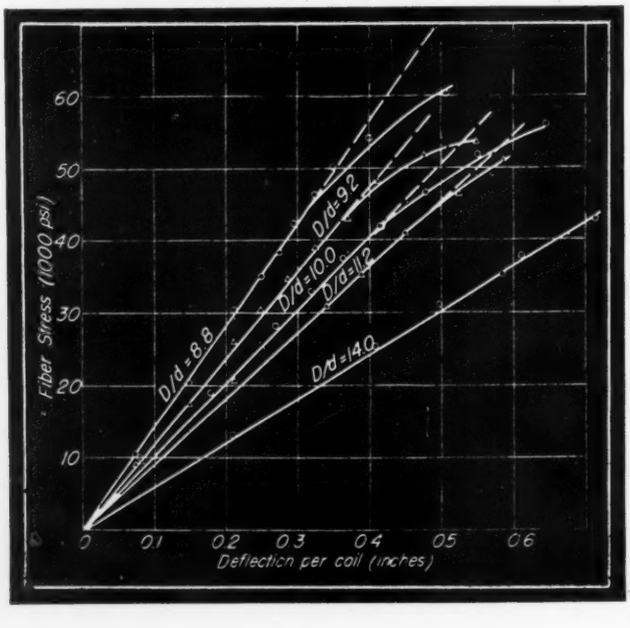
* All properties perpendicular to direction of rolling (cross-grain).
† Maximum. So specified to insure complete annealing.

typical and guaranteed mechanical properties of R303 alloy products in its various conditions of heat-treatment are listed in TABLE I.

Impressive as the mechanical properties of this alloy may be, their significance is not fully brought out until the weight factor is also taken into consideration, particularly in applications where weight conservation is of importance. A useful criterion for comparison of aluminum with the more familiar metals used in spring manufacture is the strength-weight ratio. This ratio is obtained by dividing the mechanical property under consideration (i.e., tensile strength, yield strength, or endurance limit) by the specific gravity of the material. TABLE II shows the strength-weight ratios of the aluminum alloy R303-T275 along with those of spring brass, beryllium-copper, phosphor bronze, and two grades of spring steel. The strength-weight advantage of the aluminum alloy over the other alloys becomes apparent from this table. For example, it will be noted that the yield-strength-weight ratios (sometimes called *specific yield strength*) for R303, beryllium - copper, phosphor bronze, and spring brass are 25,000, 17,700, 9200, and 7750, respectively; while the corresponding specific endurance limits are 8570, 4400, 3100, and 2980. Even against high alloy steels, the aluminum is not far behind when compared on this basis.

In addition to a weight advantage, aluminum possesses other unique characteristics by virtue of its low moduli of elasticity. Young's modulus of elasticity for R303 is 10,500,000 psi in tension, as well as in compression, and the modulus in shear, often called modulus of rigidity, is 3,800,000 psi. Consideration of the modulus of rigidity is especially important because its value governs the amount of deflection encountered in a spring. Thus, by comparison with aluminum (TABLE II), the other spring forming materials possess rigidity moduli higher by 40 to 290 percent; hence, correspondingly larger loads are required for equal deflections under otherwise similar conditions.

Fig. 2—Relationship between static stresses and deflections in helical aluminum springs



Effects 40 Per Cent Weight Saving

The fundamental equations governing the reaction of helical springs are:

$$S = K\frac{8PD}{\pi d^3} \quad \ldots\ldots (1)$$

$$f = \frac{8PD^3}{GD^4} \quad \ldots\ldots (2)$$

where S = fiber stress in coil in psi, P = load in pounds, D = diameter of coil in inches, d = wire diameter in inches, G = modulus of rigidity in psi, f = deflection per coil in inches, and K = a factor (Wahl's) depending on the ratio D/d.

If the subscript a is used to designate the associated quantities for R303 aluminum and b to designate any of the other alloys, the following relationships are easily derived:

For springs of identical dimensions it follows from Equation 1 that

$$\frac{S_a}{S_b} = \frac{P_a}{P_b} \quad \ldots\ldots (1a)$$

and from Equation 2

$$\frac{f_a}{f_b} = \frac{G_b P_a}{G_a P_b} \quad \ldots\ldots (2a)$$

Also, for similar loads in the two springs, the deflection within the elastic ranges of the materials is given by

$$f_a = \frac{G_b f_b}{G_a} \quad \ldots\ldots (2b)$$

Suppose a decidedly adverse design condition is con-

sidered in which fatigue life is the governing factor. Comparison will be made with a beryllium-copper spring as its desirable characteristics in this respect are well recognized. Although the ratio of the endurance limits of beryllium-copper to aluminum, as given in TABLE II is 36000/24000, or 1.5, it will be assumed for purposes of the forthcoming calculations that notch sensitivity of the aluminum requires a factor of safety which necessitates increasing the ratio of allowable stresses to 3; that is, $S_b/S_a = 3$.

Since $S_a/S_b = 1/3$, the maximum design load for the aluminum spring, P_a, obtained from Equation 1*a* is

$$P_a = \frac{1}{3} P_b$$

The above relationships and the respective torsional moduli when substituted into Equation 2*a* gives

$$\frac{f_a}{f_b} = \frac{6.5}{3.8} \times \frac{1}{3} = 0.565$$

Assuming further the desirability for equal deflections in the two springs, the aluminum spring will require 1/0.565, or 1.77 as many coils as the comparison spring. However, even under these assumed adverse conditions, aluminum possesses a weight advantage.

Since all other dimensions of the springs are identical, the weight of the springs is directly proportional to the specific gravity; hence

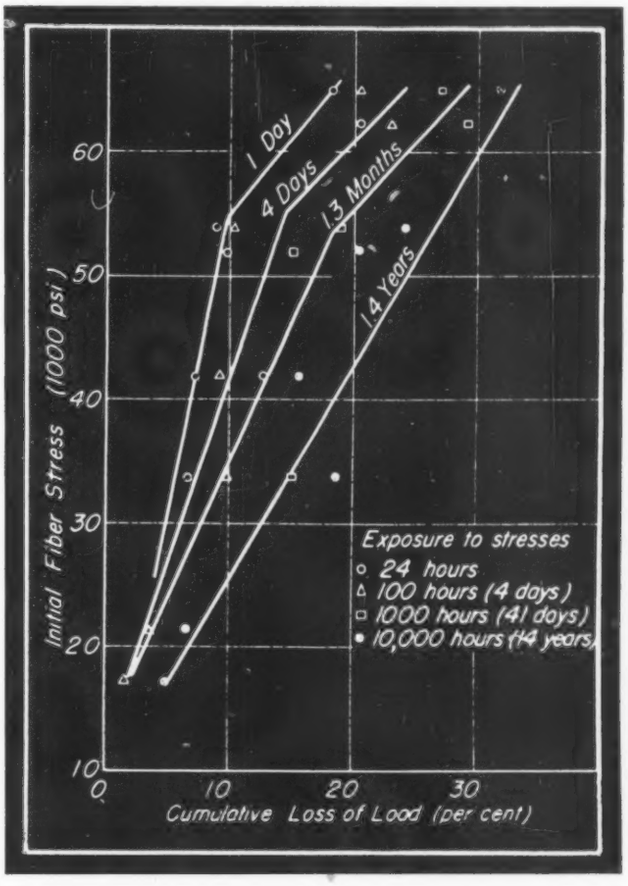
$$\frac{W_a}{W_b} = \frac{1.77}{1} \times \frac{2.8}{8.2} = 0.60$$

$$W_a = 0.60 W_b$$

in which W designates the weight.

The foregoing comparison shows that in the cited case the use of the aluminum spring effects a 40 per cent saving in weight over the copper alloy. Considerably greater weight conservation can be secured under less severe design conditions, not only with regard to the weight of the spring itself, but also because the entire reacting mechanism can thereby be lightened.

Fig. 3—Below—Loss of load in aluminum springs compressed for long time intervals. Springs were not preworked to remove set prior to test



Besides the high deflection sensitivity and light weight of R303 springs, other advantages, common to aluminum and its alloys, enhance their potentialities. Among these, good resistance to general corrosion, and the fact that the corrosion products are colorless and nontoxic can result in useful applications. Consequently the springs will not form objectionable stains when contacting fabrics in humid environments, nor will there result deleterious ef-

TABLE II—Strength-Weight Ratios for R303 and Several Spring Materials

	Young's Modulus of Elasticity		Modulus of Rigidity									
	(10^6 psi)	(ratio to Al)	(10^6 psi)	(ratio to Al)	Specific Gravity	Tensile Strength (psi)	Ratio, Tensile Strength to Specific Gravity	Yield Strength (.2% set) (psi)	Ratio, Yield Strength to Specific Gravity	Rotating Beam Fatigue Limit at 10^6 Cycles (psi)	Ratio, Fatigue Limit to Specific Gravity	Reference
Aluminum R303-T275	10.5	1.00	3.8	1.00	2.8	75,000	26,800	70,000	25,000	24,000	8,570	
Beryllium-Copper Ht. Tr. 2 hrs. @ 520 F	18.2	1.73	6.5	1.71	8.2	166,000	20,200	132,000	16,100	32,000	3,900	(1)
Beryllium-Copper Hard, Max. Ht. Tr.	18.0	1.71	6.5	1.71	8.2	193,000	23,500	145,000	17,700	36,000	4,400	(2), (3)
Phosphor Bronze Extra Spring	15.1	1.44	6.3	1.65	8.7	113,000	13,000	80,000	9,200	27,000	3,100	(2), (4)
Brass, 66-34 Extra Spring	15.0	1.43	5.5	1.45	8.4	98,500	11,700	65,000	7,750	25,000	2,980	(3), (4)
High C Steel Ht. Tr. 1575F, Drawn 940 F	30.2	2.88	10.8	2.84	7.8	225,000	28,800	179,000	22,900	80,000	10,260	(1)
Chrome-Vanadium Steel Ht. Tr. 1600 F, Draw 810 F	30.2	2.88	11.2	2.95	7.8	237,000	30,400	229,000	29,400	104,000	13,300	(1)

(1) J. B. Johnson—"Fatigue Characteristics of Helical Springs", *Iron Age*, March 15, 22, 1934.
(2) C. H. Grenall & G. R. Gohn—"Fatigue Properties of Non-Ferrous Sheet Metals", *ASTM Proceedings*, II, page 160, 1937.
(3) *ASM Handbook*, American Society of Metals, Cleveland (1939).
(4) S. L. Hoyt—*Metals & Alloys Data Book*, Reinhold Publishing Co., New York (1943).

fects from intimate contact with food and numerous pharmaceutical products.

Excellent heat and electrical conductivity, nonmagnetic and spark-proof properties are other desirable factors in certain applications.

Finally, the favorable market position which aluminum alloys have attained has in many instances resulted in lower cost per finished unit than for the more common metals.

Test Results Verify Claims

Many of the aforementioned theoretical aspects have already been verified by the results obtained with experimental R303 aluminum springs. Thus far, the helical springs which were wound for experimental examination were produced from 0.114-inch and 0.125-inch diameter wire, and were wound on various size mandrels. Wire was used in the fully heat-treated and aged T275 and T315 tempers, as well as in the freshly quenched condition. The formability of freshly-quenched R303 is excellent and, therefore, this property is utilized when severe and intricate bends are required. Where bend radii are not smaller than 4 times the wire thickness, the fully heat-treated wire may be employed, the T315 temper being slightly more ductile. In similarity to all types of spring wire, the final coil diameter depends on the temper of the wire at the time of coiling; the higher the elastic limit of the material, the greater the springback of the material.

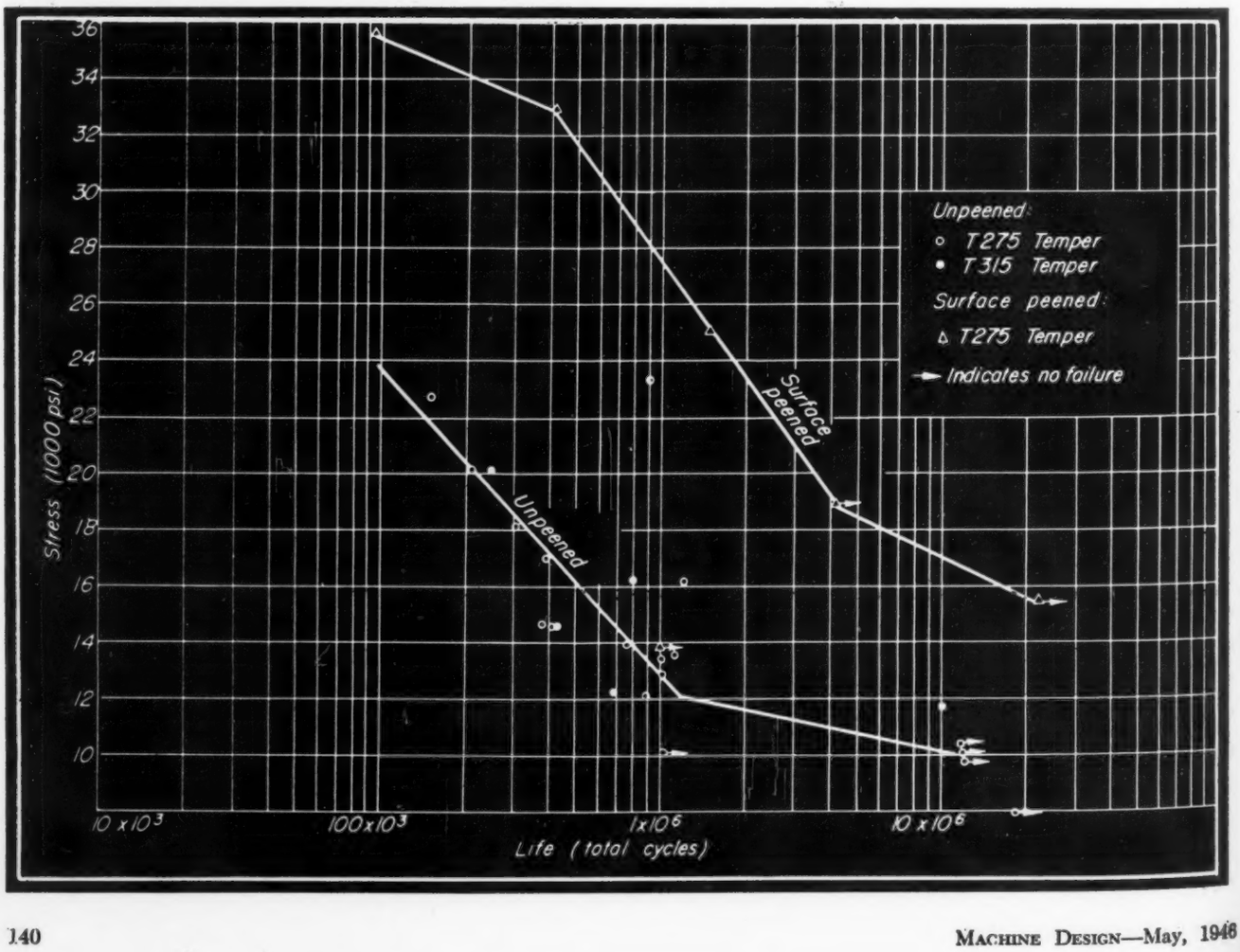
Stress-deflection characteristics for several springs of varying D/d ratios wound from R303-T275 wire are shown in *Fig.* 2. The data were obtained by measuring the deflections and the tensile loads applied to the springs. Using the observed load and the physical dimensions of the spring, the fiber stress was calculated by means of Equation 1. It becomes apparent from the linearity of the graphs that these aluminum springs exhibit under static loads a proportional limit in excess of 40,000 psi.

To determine the behavior under sustained compressive loads, eight 6-coil springs of 0.125-inch diameter wire, 1.25-inch coil diameter ($D/d = 10$), were compressed varying amounts in clamps, as shown in *Fig.* 1. After progressively increased time intervals, subsequent to release of the clamping device, the average pitch distance of the middle four coils was measured. Immediately on completing the measurements, the springs were again compressed to the initial position. The fiber stresses were computed from the initial deflections; and the after-exposure loss in distance between coils, being directly proportional to the loss in load (P, Equation 2) served as the basis for *per cent loss of load* computations. In *Fig.* 3 is shown the loss in load by springs that were subjected to sustained compressive stresses. Since none of these springs were preworked after winding, the permanent set was higher than would be encountered if such procedure were followed.

An example of the reduced permanent set in preworked

(*Concluded on Page* 178)

Fig. 4—Fatigue life of R303 aluminum springs under repeated compressive loading



Trends in Automotive Transmissions

By Colin Carmichael
Associate Editor, Machine Design

Part II

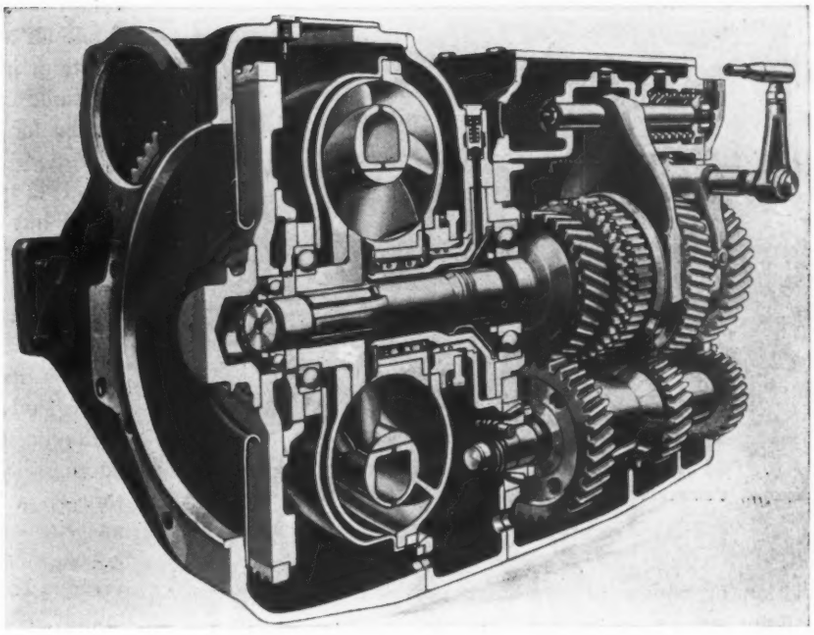
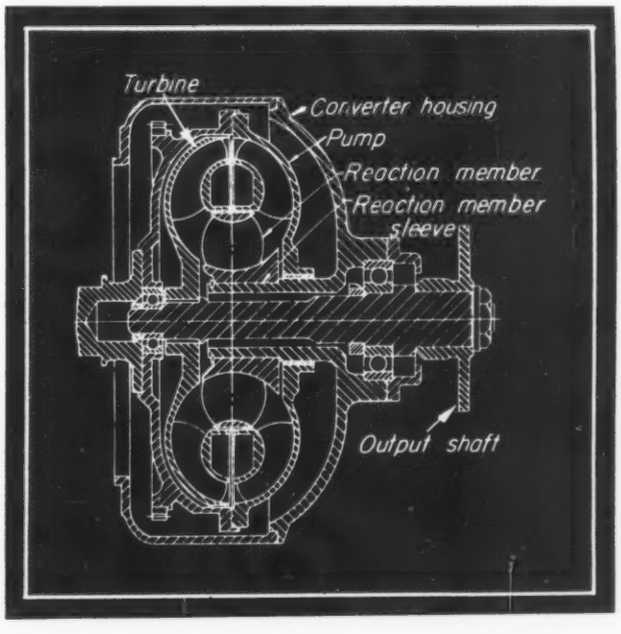


Fig. 5—Above—Cutaway view of White Hydro Torque drive consisting of single-stage hydraulic torque converter and automatic two-speed transmission

Fig. 6—Below—Schematic diagram of single-stage hydraulic torque converter of the type shown in Fig. 5



FOR MOST satisfactory all-round performance an automotive transmission must be carefully designed to match the characteristics of the engine and the expected service of the vehicle. In last month's article it was shown that the ratio of engine speeds at maximum horsepower and maximum torque furnishes a guide to the ratios of a selective-speed transmission. With such ratios, a 3-speed gearbox furnishes adequate low-gear wheel torque for the most adverse conditions of load, gradient or road surface. Additional speed changes are a refinement which can improve the economy of operation under certain conditions and make possible a fuller utilization of the capacity of the engine.

On the other hand, busses and trucks which have a much lower ratio of engine power to vehicle weight require a wider range of speed changes to insure adequate top speed coupled with good low-speed acceleration and pulling power for heavy loads and gradients. With step changes roughly equal to the ratio of engine speeds at maximum horsepower and maximum torque, more steps are therefore necessary. For example, if the ideal ratio is 1.74 per step, a five-speed gearbox would furnish ratios of 1, 1.74, 3.06, 5.36, and 9.4. The alternative of retaining the three-speed gear-box with the extended range would give ratios of 1, 3.06 and 9.4 but would necessitate operation at certain speeds in the region where engine torque decreases with decreasing speed. Should this condition be encountered on a heavy grade at full throttle, it will be evident that the drop in torque corresponding to a drop in speed would result in a further drop which in turn would quickly reduce vehicle speed—the opposite of good "lugging" ability. Until vehicle speed had dropped substantially, a downshift would result in overspeeding the engine, hence there would be certain speeds at which performance would be poor.

In heavy trucks, even five ratios do not always furnish top performance in hill climbing and fuel economy. Pro-

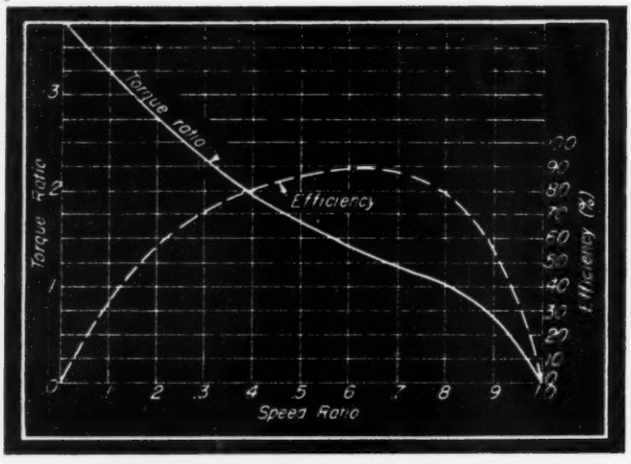


Fig. 7—Performance curves for single-stage hydraulic torque converter of similar design to that shown in Fig 6.

vision of additional ratios, for practical reasons, necessitates some form of auxiliary gearbox, with consequent complication in the operation of shifting. A neat solution, known as the Mono-Shift, has been devised by Mack, *Fig.* 8, in which special ratios are employed in the main five-speed transmission so that with a two-speed auxiliary unit an even progression of ten speeds is obtained. A single gearshift lever controls all shifts, the five main speeds being obtained in the conventional manner while the auxiliary or "compound" set is power shifted in response to movement of a small flipper located in front of and just below the gearshift lever knob. The flipper actuates a spool-type selector valve controlling a vacuum cylinder. An over-riding master valve controlled by the clutch pedal is also employed to insure that the compound shift shall take place only when the clutch is released.

Instead of a multiple-speed gearbox there is a trend toward the use of torque converters, *Fig.* 5, providing infinitely variable ratio changes between certain limits While all automotive transmissions are torque converters, the term in general usage seems to be restricted to the hydraulic variable-ratio type. Although production of hydraulic torque converters so far has been limited to bus aplications, their performance characteristics are eminently suited to passenger-car requirements. Before covering actual bus transmissions, therefore, the torque converter will be discussed with reference to its possible use in smaller cars where the requirements are less stringent.

In its simplest form the hydraulic torque converter consists of three simple elements—a pump, a turbine, and a reaction member, *Fig.* 6. The pump element, which corresponds to the impeller of a centrifugal pump, delivers hydraulic fluid to the turbine runner, in which the fluid flows radially inward. Fluid leaving the turbine flows through a series of guide vanes in a stationary reaction member and again enters the pump. This stationary reaction member is an essential feature of a torque converter. It absorbs the difference between the input and output torques, and without it no change in torque could be effected.

Performance of a torque converter such as shown in *Fig.* 6 is best represented by a pair of curves for efficiency and torque ratio plotted against speed ratio, *Fig.* 7. These

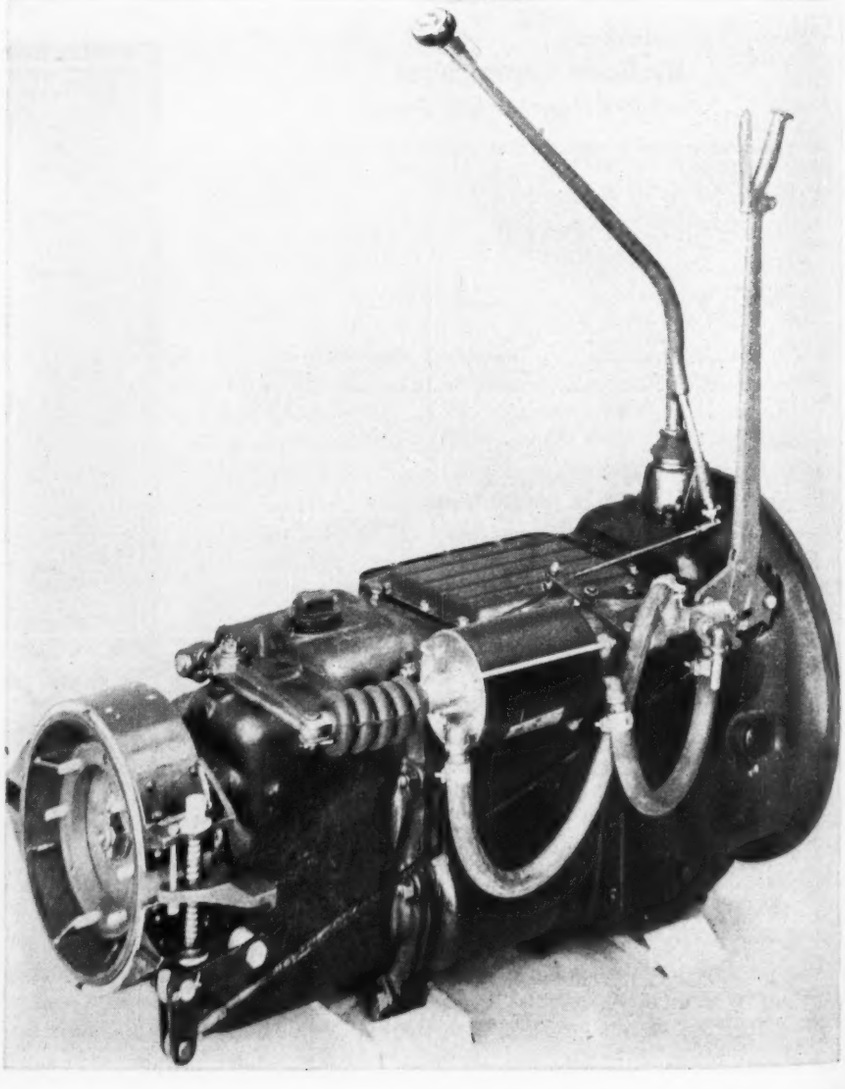


Fig. 8—Right—Mack Mono-Shift transmission for heavy trucks employs 5-speed main gearbox with auxiliary 2-speed unit, giving 10 speeds in all. Auxiliary, or compound, shift is effected by vacuum controlled by flipper located under knob of gearshift lever—Photo, courtesy Mack-International Motor Truck Corp.

three quantities are related by the equation

$$\frac{T_2}{T_1}=\frac{N_1}{N_2}\times\frac{e}{100} \quad (1)$$

where T_2/T_1 is the ratio of output to input torques, N_2/N_1 is the ratio of output to input speeds and e is the efficiency. These curves, whose shape depends on the design, vary slightly in shape at different input speeds but a single set of curves of the type shown in *Fig.* 7 is sufficiently accurate for performance analysis of the particular converter design to which they apply. The actual torque absorbed is proportional to the square of the input speed and to the fifth power of the converter diameter.

These characteristics are well suited to the automotive drive as a little consideration will show. The stalled torque ratio (at zero speed ratio) for the converter represented by *Fig.* 7 is 3.75, which is comparable to the low-gear ratios used in passenger cars. The converter would be designed to absorb the maximum torque of the engine at the speed at which maximum torque occurs—2100 rpm in the example of *Fig.* 1 in last month's article. At an idling speed of, say, 400 rpm the torque-absorbing capacity would be only $(400/2100)^2 = .036$ or 3.6 per cent of the maximum, hence at idling speed the torque transmitted would be barely sufficient to propel the car. This obviates the need for a clutch, and idling the engine is virtually equivalent to going into neutral. However, when it is desired to get under way it is only necessary to speed up the engine to the point where substantial torque can be absorbed. The torque ratio of 3.75 at zero car speed provides ample multiplication and a quick, smooth start results.

As car speed picks up, the speed ratio increases and the torque ratio drops, as shown by the curve, *Fig.* 7. This is equivalent to an infinite number of minute upshifts which occur automatically in response to speed and load conditions. Actually, the converter at all times is attempting to balance Equation 1. Any unbalance results in acceleration or deceleration of the vehicle or engine or both until equilibrium is established in accordance with the performance characteristics in *Fig.* 7.

When speed ratio reaches 0.81 the torque ratio is 1. This is called the "clutch point" because at this condition the converter acts merely as a coupling or fluid clutch; there is no torque conversion and no torque reaction on the reaction member. If speed ratio were allowed to exceed that at the clutch point the torque ratio would drop below unity, torque reaction on the reaction member would be negative and the converter would act as a brake. In automotive applications, therefore, when it is desired to operate at speed ratios up to 1.0 it is necessary to do one of two things:

1. Cut loose the reaction member so that it is free to revolve with the fluid and exerts no drag
2. Shift into direct drive, by-passing the converter.

Both expedients are in current use, the first being employed in the White Hydro Torque drive, *Fig.* 5, and the second in the GM Coach Hydraulic Drive. By cutting loose the reaction member the converter becomes simply a fluid coupling for which the torque ratio is always 1 to 1. The changeover from converter to coupling and vice versa is automatically effected in the White drive by simply mounting the reaction member on an overrunning clutch or "free-wheel" unit, *Fig.* 9. Since at the clutch point the direction of torque reaction tends to change, the free-wheel unit makes it impossible for the reaction member to suport negative torque.

Performance curves for a converter-coupling combina-

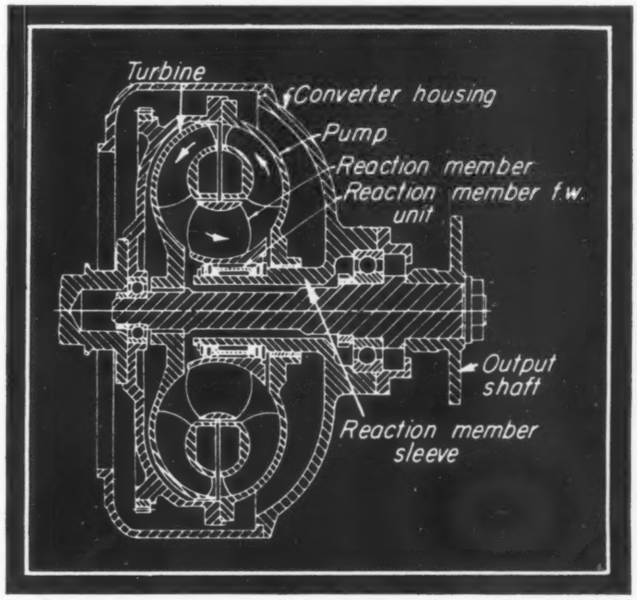
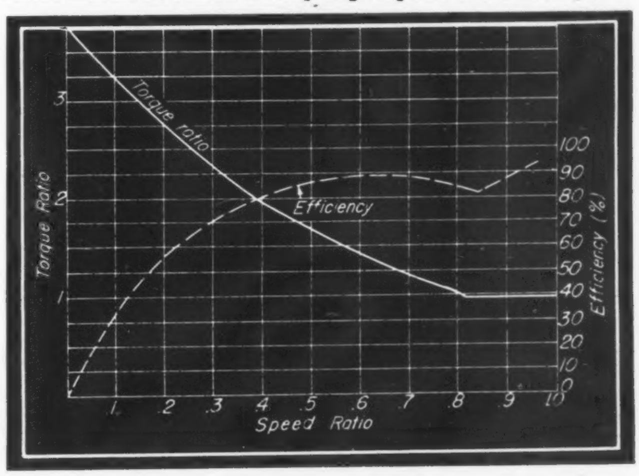


Fig. 9—Above—Schematic diagram of single-stage torque converter with reaction member mounted on free-wheeling clutch. At speed ratios above clutch point the converter becomes a coupling due to free rotation of reaction member

Fig. 10—Below—Performance of single-stage torque converter which becomes a coupling beyond the clutch point



tion of this type are shown in *Fig.* 10. Inasmuch as the torque ratio of a coupling can only be 1 to 1 the efficiency, e, from Equation 1, necessarily is equal to the speed ratio, N_2/N_1, times 100. Although this would indicate 100 per cent efficiency at 1-to-1 ratio, the torque-transmitting capacity, which is a function of slip between the driving and driven elements, would be zero hence the point has no practical significance. The curves therefore are shown as stopping just short of the maximum.

By the alternative of shifting into direct drive at the clutch point and bypassing the hydraulic unit, the effi-

ciency jumps to 100 per cent as shown in *Fig.* 11. Actually, of course, it would be impossible to operate in the range between the clutch point and 1 to 1 speed ratio and in effecting the shift the engine must be brought up to the speed of the driveshaft. The slight pause while this is being done has a negligibly small effect on the acceleration of the vehicle, as shown by the actual accelerometer record in *Fig.* 12. The shifting of the G.M. Coach Hydraulic drive is done automatically and the driver has only to select the direction of motion—forward or reverse—and then control the speed solely by the throttle and brake pedals.

Design of the hydraulic torque converter itself has an important bearing on its application to vehicle drives. Thus, the maximum torque ratio and the speed ratio at the clutch point are controlled by the design proportions. Units now available are of two distinctly different designs: The single-stage unit, *Fig.* 5, 6 and 9, developed by Adolf and Heinrich Schneider and used by The White Motor Co.; and the well known single-pump 3-stage turbine unit originally developed by Alf Lysholm in Sweden and subsequently improved for bus service in this country by the Spicer Mfg. Corp. With a maximum torque ratio as high as 6 to 1, the multi-stage converter is suitable for bus drives without an auxiliary transmission if provision is made for going into direct drive.

While the single-stage unit is much simpler in construction, its more limited torque ratio is not quite adequate for a heavy bus—although it would be for a passenger car. To extend the range, White has combined the torque converter with an automatic two-speed transmission, resulting in a completely automatic drive with full-throttle performance somewhat as shown in *Fig.* 13. The shift from low gear to high gear and vice versa is controlled by the relative speed of the input and output shafts, which is a measure of the torque requirements.

Referring to *Fig.* 13, the dotted portions of the torque ratio and efficiency curves indicate the performance if only high gear were employed. To avoid these low efficiencies at low speed ratios, the two-speed transmission is shifted into low at a predetermined speed ratio. With the torque multiplication now taken care of by the gear, the converter functions as a coupling down to the clutch point, when it then takes over the job of providing additional torque multiplication.

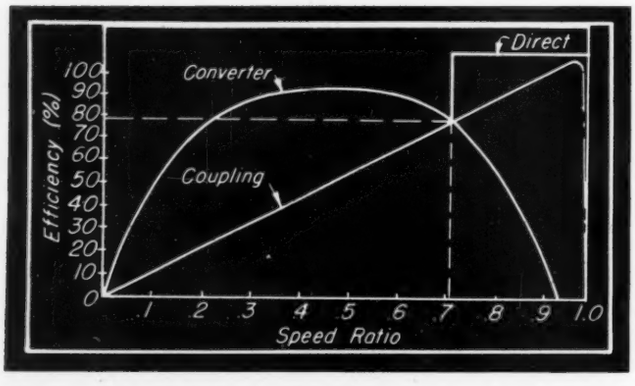


Fig. 11—Above—Efficiencies of torque converters with change to direct drive and with change to coupling

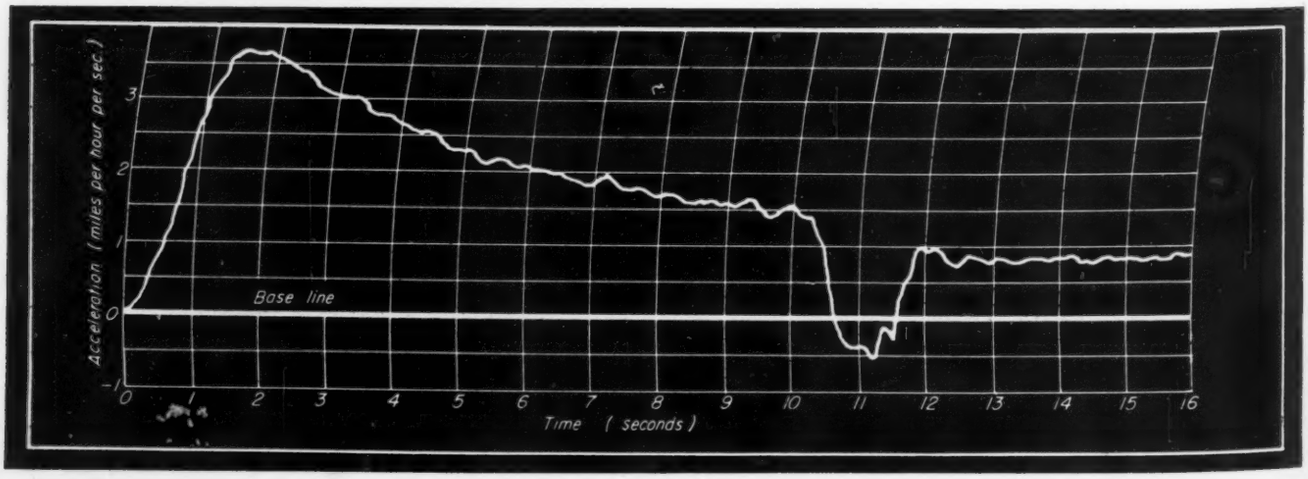
Fig. 12—Below—Acceleration of 41-passenger city bus with a hydraulic transmission which changes to direct

Differential Gear Controls Shift

Control of the shift is effected by a differential gear mechanism. The differential cage is driven in proportion to the engine speed while one of the pinions rotates in proportion to the converter output speed. The other pinion, therefore, rotates at a speed in proportion to the difference between input and output speeds and is used to drive a governor with two sets of weights which, in turn, actuate two switches for effecting upshift and downshift respectively.

When, due to increase in speed ratio resulting in a smaller speed difference, the differential governor closes the upshift switch the following sequence of events occurs: First the engine speed is decreased by automatic closing of the throttle while air pressure is admitted to a shift piston in the transmission cover, forcing the shift collar from low toward high; then, when the synchronizing clutch establishes the proper speeds, the shift is completed and the throttle opened automatically to the point corresponding to accelerator pedal position. During the shift no reaction is felt on the accelerator pedal.

Downshift is accomplished in response to a signal from the downshift governor switch when the speed difference reaches a predetermined value. First the throttle is momentarily closed while air pressure is shut off from the shift piston, allowing a spring to move the shifter fork to-



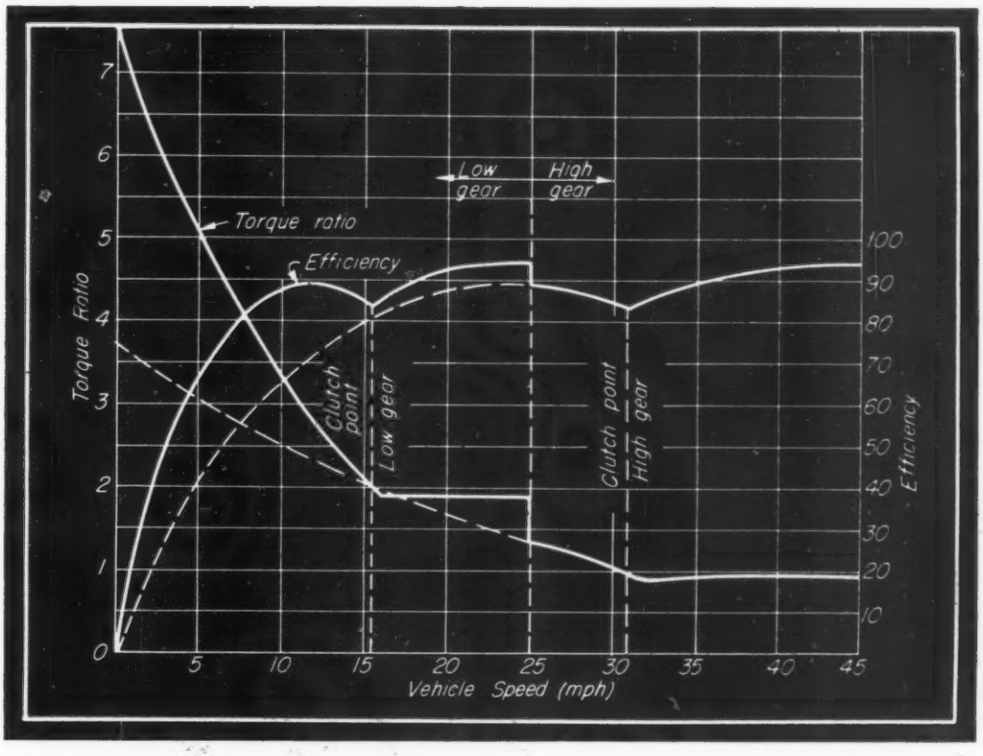


Fig. 13—Efficiency and torque available at various vehicle speeds for a transmission of the type shown in Fig. 5

ward low gear. As soon as the synchromesh collar is disengaged from the main drive gear, the throttle automatically opens again to speed up the engine. When synchronization is established the ignition is interrupted momentarily so that final engagement to low gear is made under no transmitted load. On complete engagement, ignition is restored. The foregoing sequence of operations will be recognized as corresponding to the familiar "double-clutching" procedure necessary with a standard transmission. In this case the series of operations is employed to relieve the synchronizing mechanism, thus insuring longer life for this unit. In the absence of a clutch, the engine speed is automatically manipulated to give the same effect.

Regardless of actual speed ratio, below a certain vehicle speed the transmission automatically goes into low through the action of a low-speed governor driven by the output shaft of the transmission. This governor contains a switch which prevents the occurrence of an upshift following a power downshift, giving the driver a means of staying in low gear, irrespective of torque requirements, until he decides to shift up. This he does by increasing the speed of the vehicle. It also prevents "hunting" when the transmission is operating under conditions where torque requirements could be met in either high or low gears.

Even in descending a hill, when the vehicle drives the engine, the characteristics of the converter give improved performance. Inasmuch as the vanes of the pump and turbine wheels have not been designed for transmitting power efficiently in the reverse direction, the converter efficiency is lower. Due to the loss of power there is an increased braking effect, while the engine is driven at a lower speed than it would be with a mechanical transmission, resulting in less engine wear. Because of the low efficiency in descending a hill it is not possible to establish the minimum speed difference required for an upshift, hence the transmission will stay in low gear regardless of vehicle speed.

It has been predicted* that the passenger car automatic transmission of tomorrow will include a hydraulic torque converter in conjunction with a manually selected mechanical gear to be used for coasting down long hills or for long upgrade pulls, the torque converter on such occasions being locked out. The coasting gear may be necessary to comply with the law in certain states, and if it has to be there it might as well be used to provide torque multiplication for hill climbing.

Inasmuch as hydraulic torque converters so far have been built in relatively small quantities for high-priced vehicles and for military applications such as tanks and tank destroyers, low-cost mass-production methods have not been used in their manufacture. Consequently they are quite expensive. A big job of redesign remains to be done and production runs must be quite large if a torque converter competitive with the conventional gear transmission with or without a fluid coupling is to be produced. It is possible that the answer may lie in the precision casting techniques recently developed on an important scale. By such methods the necessary special shapes of vanes can be produced with sufficient accuracy and surface finish without resorting to hand work. If the demand is sufficient it may be possible to employ assemblies of stampings. In any case, the torque converter presents a challenge to the cost-conscious production engineers*.

In the preparation of this article much of the information has been obtained from the following group of papers presented at recent meetings of the Society of Automotive Engineers:

1. "Torque Converters?"—Raymond J. Miller, SAE Annual Meeting, Detroit, Jan., 1946.
2. "Hydraulic Transmissions for Motor Vehicles"—A. H. Deimel, SAE National War Material Meeting, Detroit, June, 1944.
3. "Hydraulic Transmission Performance"—A. H. Deimel, SAE Annual Meeting, Detroit, Jan., 1946.
4. "The White Hydro Torque Drive"—R. Brunken, SAE Meeting, Detroit, Jan., 1946.

*See Reference 1 at end of article.

Lost Wax Investment Castings

DESIGNERS have long been intensely interested in the "lost wax" or "investment casting" process as developed for the mass production of machine parts from heat and corrosion-resistant materials not readily machined or formed. This pictorial presents the primary steps required in the production casting of precision parts as practiced at the Kokomo, Indiana, plant of the Haynes Stellite Company.

Through developments brought about by cooperation between Haynes Stellite and the Union Carbide & Carbon Research Laboratories, Inc., the process opens to the machine designer an unlimited field of possibilities for utilizing intricate contours, airfoils and shapes hitherto considered impossible or impractical. Now he can design for maximum efficiency by adapting and combining desired sections, however intricate, and specifying the necessary super alloy for the job.

1.—From a drawing of the part to be produced, above, a "master pattern" is made using brass, steel, Stellite alloy, wood, or other suitable material, depending upon the design of the part. This pattern is a replica of the desired piece, sufficiently oversize to allow for shrinkages.

2.—Using the master pattern, injection dies are prepared from a suitable soft metal. The first step, right above, shows master pattern being imbedded in plaster in order to make the top half of the die.

5.—These patterns are inspected as they move along a conveyor belt line. Special blind risers are wax welded to specific places on the pattern, determined by tests, to assure sound castings and compensate for shrinkage. Above, operator is mounting two such wax patterns on a main gate with wax runners.

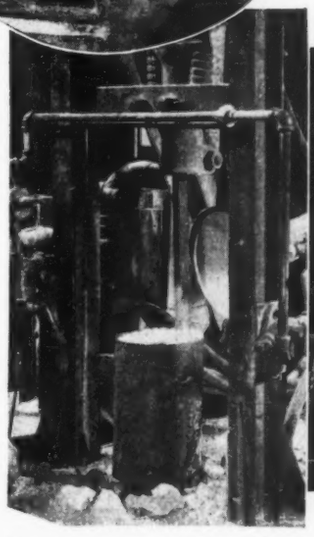
3.—With top half of the die completed, plaster is removed and the second half is poured, above.

4.—Necessary cores are placed and, with the dies closed, melted wax is injected into the die cavity by means of a pneumatic ram. The wax replica of the master pattern thus formed is removed from the die, right. Wax gate is manually broken off.

Steps in Producing Precision Parts

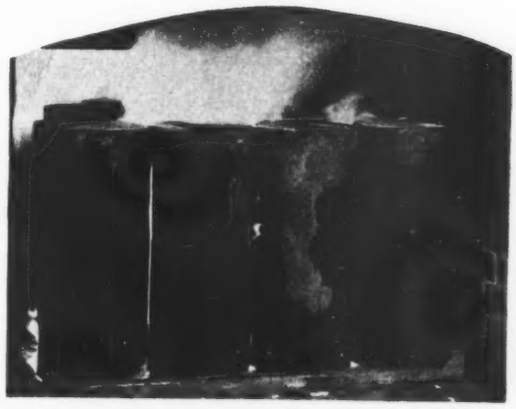
6.—Patterns, with gate and risers assembled, are mounted on wax hubs especially molded for the purpose, above.

7.—Pattern assembly is then dipped in fine silica suspended in a suitable medium, left below. This material lines the mold after the wax pattern is melted out providing a smooth finish on the castings.



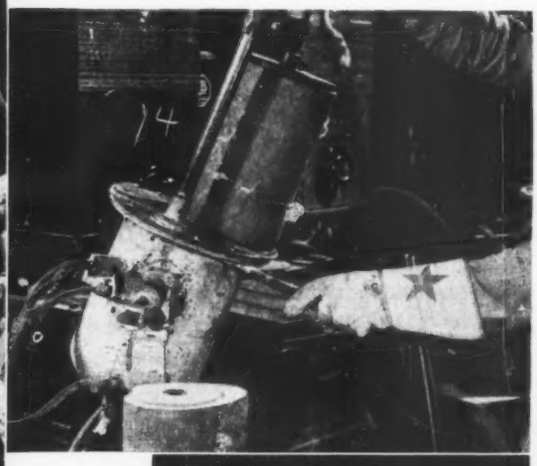
8.—Following this dip the pouring end of the wax assembly is mounted on a steel plate and the remaining area sprayed with silica material, below. Coarser grains are "stuccoed" to the dip coating and the assembly dehumidified in a drying tunnel. A flask of Hastelloy C is then sealed to the steel plate with wax.

9.—Into the flask and around the prepared pattern assembly is ladled a chemically hardening investment material, below. Filled molds are vibrated for about one hour to pack the material and eliminate air pockets.



10.—Mold is cut to length and allowed to age for several hours. Mouth down, molds are passed through a furnace where the pattern is melted out and all traces of wax burned away at from 1300 to 1900 F, above.

11.—Hot, baked mold, clamped in position over a small electric furnace, below, is filled by inverting entire furnace. Application of air pressure assures sound, dense metal as well as complete filling of light walls or thin edges.

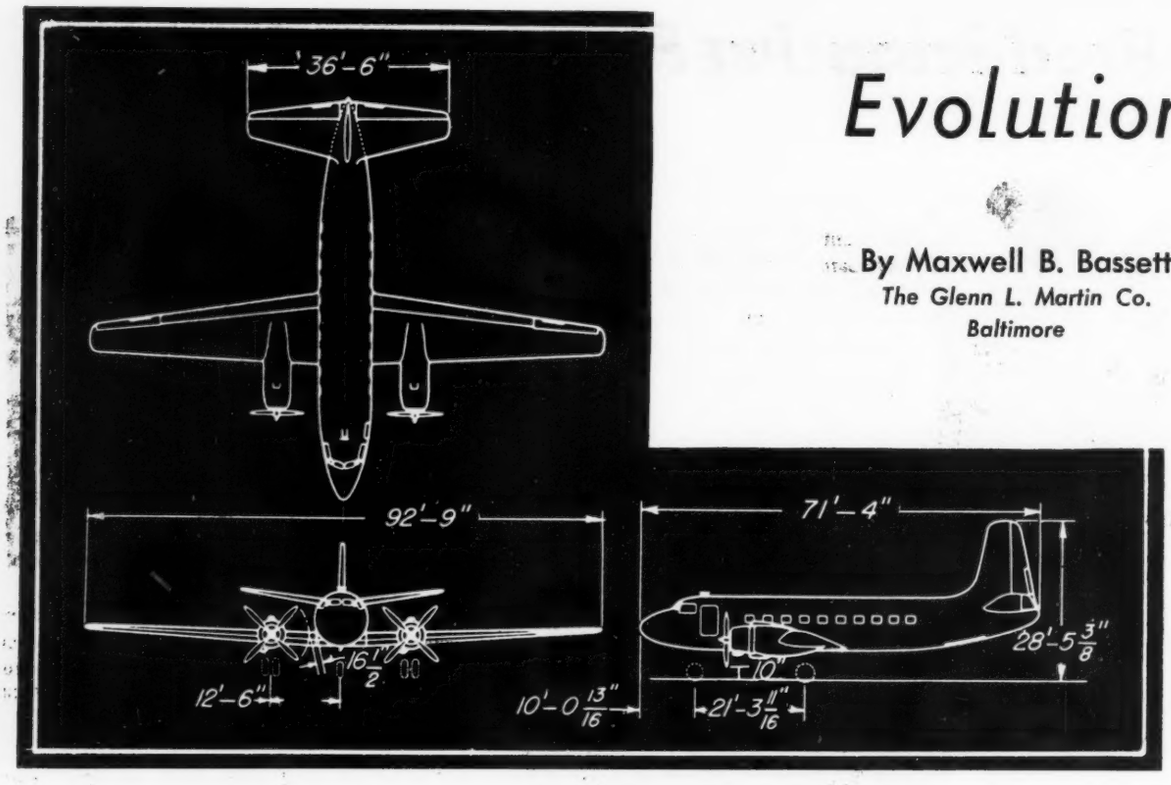


12.—After about four hours cooling, casting assembly is knocked from the mold by a pneumatic hammer, above. Gates and risers are removed by means of abrasive cut off. All but fragile castings are then shot blasted to remove scale and remaining traces of investment.

13.—Gate areas and surface irregularities are removed and smoothed by grinding. Castings are then sand blasted to perfect surface finish. Passing dimensional inspection, parts are subjected to Zyglo fluorescent test, right. For final check on quality of critical parts, x-ray is used to detect internal imperfections.

Evolution of M

By Maxwell B. Bassett
The Glenn L. Martin Co.
Baltimore



SOME years ago it began to be apparent that a need was arising for a new transport airplane to replace the "workhorses" of the airlines—those airplanes operating over the major trunk lines of the United States and other countries. An important question that had to be answered before even a preliminary new airplane design could be laid down on the drafting board was: How far do most air travelers fly? It was found that 69 per cent of air travel was under 500 miles, 12 per cent from 500 to 800 miles, and 19 per cent more than 800 miles. Thus, preliminary work established the necessity for providing a new airplane for ranges up to 500 miles with low-cost operation, higher speeds, improved passenger comfort, increased seating capacity and, of course, a high degree of safety.

Many airlines of the United States were visited in early 1943 by representatives of the aircraft manufacturer for conferences on preliminary design, and there began a long period of constant co-operation to evolve the new transport airplane known as Model 202, *Fig.* 1. To the airlines must go much credit for their splendid co-operation, constructive criticisms, design ideas and other contributions without which this ship could not have been designed.

Early conferences revealed a difference of opinion on certain features desired. Some believed that the new airplane should be fast, others thought it should be slow for economy. Some preferred high-wing types, others, low-wing designs, and still others could not agree on size.

More than twenty-five airplane designs were studied by the manufacturer before the final basic airplane configuration, power plant size and other major design features were determined. All kinds of unconventional arrangements employing uni-twin engine installations, pusher propellers, engine extension shafts, etc., were studied, always keeping in mind safety, economy and passenger comfort. Of special importance were studies to determine the

From a paper presented at the recent national aeronautic meeting of the Society of Automotive Engineers, in New York.

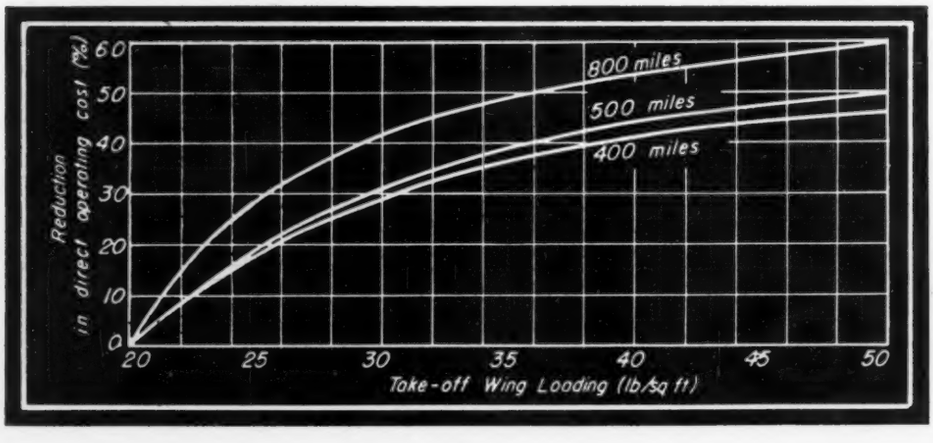


Fig. 1—Top—Leading proportions of the new Martin Model 202 transport plane, which was designed for service on major domestic and foreign trunk lines

Fig. 2 — Left — How operating cost of a two-engined transport plane is reduced by increasing the wing loading, for various trip distances

f Modern Transport Airplane

economical number of engines, best combination of power loading and wing loading, and whether to use a high or low-wing design.

Since twin-engined airplanes had established remarkable safety records and proved themselves satisfactory for medium ranges, economy was the criterion chosen for deciding the number of engines. After making and studying designs having two, three and four engines, it was found that the twin-engine design was unquestionably the most economical for the following reasons: (1) Less engine weight per installed horsepower, (2) less overall airplane drag, (3) lower airplane first cost, (4) shorter time for engine run-up, starting, and airplane maneuvering, (5) less equipment and control weight, (6) higher cruising speed with more economy than an airplane with more engines when both types were designed for the same operating conditions to meet Civil Airworthiness requirements.

Intimately tied in with airplane cruising speed are wing loading and size of power plant, which were subjects for extensive research to determine the most economical combination. Many engineers and airline personnel argued that a slow airplane with small engines and large wings would be most economical while there were those who were equally convinced that a fast airplane could be obtained at no extra cost, with higher wing loadings.

Two separate and distinct approaches were taken to secure the answer to this problem. The first method was to make a series of actual preliminary designs employing different size power plants then available. The airplanes were designed to carry a fixed payload over average ranges of 400 to 800 miles.

Results of this first study showed that the faster airplanes with higher wing loading were most economical and that the only limit to wing loading was the high-lift device which could be developed to meet the then existing limit of 80 mph for stall speed. Landing distance also was found to become critical at higher stall speeds. For example, the landing distance required at a stall speed of 80 mph was approximately 3750 feet.

The change in operating cost with wing loading is shown in *Fig.* 2. Some of the factors which result in lower operating costs for the airplanes with higher wing loading are higher speed and reductions in overall drag, in design load factors, in wing and tail weights, and in gross weight of the airplane to do a given job.

Most airline personnel agreed that the faster airplane would have lower operating costs and would be generally most desirable. However, because some skepticism still existed, a second study attacked the problem from a slightly different angle.

Instead of using actual power plants and designs, a comprehensive study was made assuming values for the weights of power plants, wings, etc., thus eliminating variables resulting from use of power plants of different manufacturers. A theoretical power plant just large enough to meet the operating criteria was assumed to be installed in airplanes with various wing loadings.

How Operating Cost Is Affected by Speed

Since the theoretical airplanes were designed to carry a fixed payload 300 miles, the fuselage space was held constant. However, gross weight varied with wing loading, power loading and fuel. Cruising speed was allowed to vary for all airplanes and was computed at 60 per cent of normal rated power. As in the original study, certain criteria such as enroute ceiling and airport altitude were held constant.

As shown in *Fig.* 3, the lowest operating costs occur for these airplanes having cruising speeds between 255 and 260 mph at 10,000 feet altitude using 60 per cent of normal rated power. The twin-engined airplane under discussion is shown on the curve (M-202). *Fig.* 4 is a plot of per cent changes in operating cost versus wing loading. This curve indicates that the higher wing loadings, of approximately 40 to 50 pounds per square foot, are most desirable. As mentioned before, the practical limitation to the wing loading is the maximum lift coefficient that can be obtained for landing to permit use of airports with reasonable runway lengths. Model 202 is shown on *Fig.* 4 at a wing loading of 44.5 psf.

It should be pointed out that cruising speed at which minimum operating cost occurs is only true for the particular set of conditions under consideration and at the present state of the art. For example, if at some later

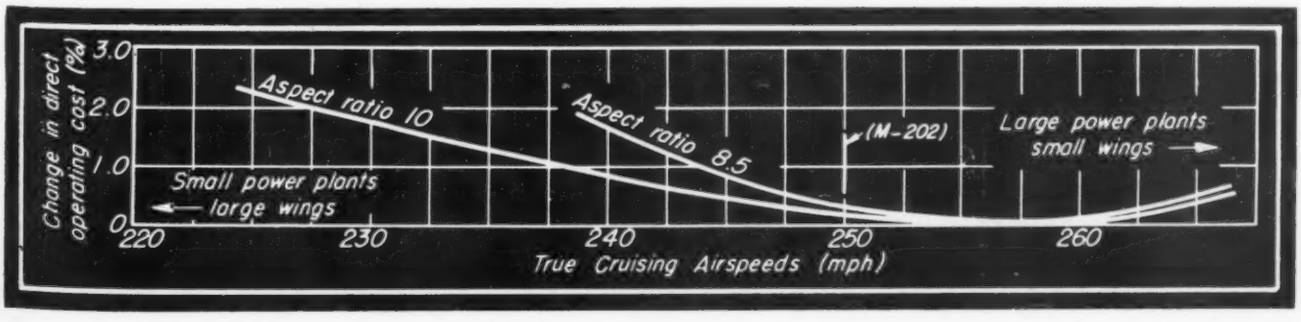


Fig. 3—Effect of cruising speed on operating cost of a two-engined transport, cruising at 10,000-ft altitude on 60 per cent of take-off power, for a 300-mile range

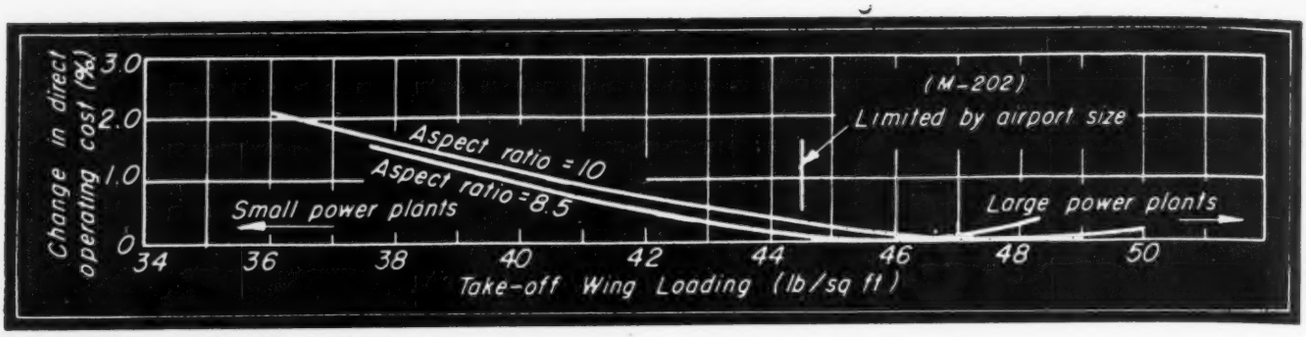


Fig. 4—Effect of take-off wing loading on operating cost of two-engine transport for same conditions as in Fig. 3

date it were possible to obtain lower airplane drags, higher cruising powers per pound of installed engine weight, more than 60 per cent of normal rated power for cruising, lower fuel consumptions or similar improvements, the cruising speed for minimum operating cost would be higher than those indicated in this study. This is the reason that the transport of 20 years ago had cruising speeds of only 100 mph instead of the 250 to 300 mph available today.

Wherever engineers gather to discuss transport airplanes, a favorite subject is the relative value of a high-wing airplane versus the low-wing type. Needless to say, every airline operations man has at one time or another sworn that his airline will purchase only the high-wing type where passengers can take one step into the cabin and, when seated, have clear vision toward the ground without glare from the wing. It must be admitted that these advantages are very real, but the subject is a complex one and must also be examined from an engineering and economic viewpoint.

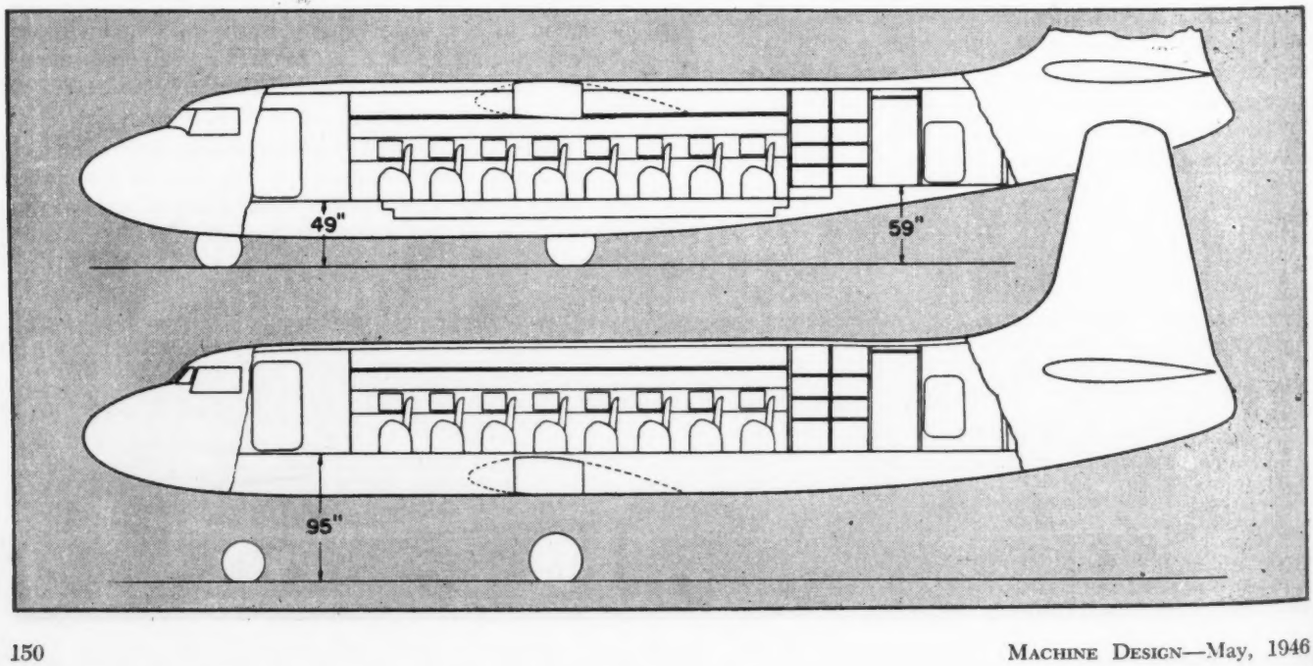
Since the airlines were so divided on their opinion as to the relative merits of low and high-wing types, every conceivable arrangement of both types was designed. The low-wing type had the very real advantage of a greater percentage of useful load and consequently more payload-carrying ability. The structural design is simpler since the main landing gear struts are shorter and the payload is being carried on top of the wing structure instead of suspended from the wing. The fuselage of the low-wing airplane is lighter, cheaper and simpler to build because no added protection is required in the belly to protect passengers and equipment in case of inadvertent wheels-up landings. In a wheels-up landing of the low-wing type, wing structure and nacelles absorb most of the shock.

Numerous arrangements of the high-wing type were designed to find the best compromise. Arrangements of the gear folding into the nacelles were heavy because of the length of the gear. Although their length could be reduced by supporting the gear from, and retracting it into, the fuselage, this required a longer, heavier fuselage. It also had the disadvantage of cutting the cabin in half, resulting in inefficient space utilization. With the main gear retracting into the nacelles or wing, length could be saved by allowing the wing spars to be cut at the fuselage and the loads carried around heavy bulkheads.

After a study of all possible arrangements, the best all-around high-wing design was considered to be that shown in *Fig.* 5, which had the most functional cabin and least weight penalty. Also shown in the same figure is a comparable low-wing design. The difference in weight empty and payload is approximately 720 pounds in favor of the low-wing type.

After determining the differences in payload and speed of a low and high-wing design, the final measuring stick was total operating cost. It is quite apparent that it

Fig. 5—Comparison of cabins for high-wing and low-wing airplanes. Low-wing design weighs about 720 lb less



would take more of the high-wing type airplanes to move a certain number of ton miles or passenger miles per year if the ground loading time of both types of equipment were the same. Since ground handling time and costs might be considerably lower for the high-wing type, a detailed cost comparison was made.

This study covered cost of loading equipment, loading times determined from studies on mock-ups, extra personnel at each airport, and the weight and speed differential between the two airplanes. Neither airplane was considered to carry built-in loading equipment.

A study was made to combine all these factors and determine how much more time could be spent in loading the low-wing airplane at each stop before the total costs of both types would be equal. Obviously, the more schedules flown through each station per day and the longer the average hop length, the less would be the penalty imposed on the low-wing airplane.

In *Fig.* 6 is shown a plot of break-even time or extra time above a 5-minute minimum loading time assumed for each airplane, which could be allotted to the low-wing design for loading versus trip length. This is plotted for several one-way schedule frequencies per station per day. It can be seen from the curve for an airline stopping 10 times per day per airport on a system with average trip lengths of 200 miles, that 14½ minutes longer could be spent loading a low-wing airplane than a high-wing design at each stop before the total operating costs of both types would be equal. Mock-up tests and actual observation reveal that in use of automatic loading equipment plus the extra personnel assumed in this study for the low-wing airplane, the difference in loading time is nowhere near 14½ minutes. This leads to the conclusion that the total operating costs of the low-wing design are less than for a high-wing type.

Why Low-Wing Design Is Superior

The low-wing configuration finally selected for the new twin-engined airplane has the following advantages: First, it has greater useful load and lower total operating cost; second, integral automatic passenger ramps and loading equipment can be designed for the low-wing airplane for much less than the 720 pounds necessary to build a comparable high-wing design; thirdly, it has lower first cost.

Front and rear automatic integral loading ramps have been designed for the low-wing airplane for a total weight penalty of only 143 pounds. With the shorter main loading ramp for passengers and no forward ramp for the cargo compartment required on a high-wing design, the weight penalty on this airplane would be 80 pounds. It is, therefore, apparent that integral loading ramps could be installed in the low-wing airplane for a net weight penalty of only 63 pounds—a much smaller weight penalty than the 720 pounds basic differential.

Passenger visibility is much better in the newly designed ships because of the relatively narrow wings and, consequently, fewer passengers over the wing. Also, the glare problem is approaching solution through the development of anti-glare finishes.

The resulting preliminary design with low wing and two engines each producing 2100 bhp for take-off is shown in *Fig.* 1. It was designed to get into and out of 3750-foot airport runways and to comply with the 80-mph maximum stall speed. Operating range was approximately 700 miles with full payload of 30 passengers and cargo.

At this stage in the design, airlines wanted a combination passenger-cargo airplane, but it was recognized that later requirements might call for more passenger seats and less cargo. The fuselage was, therefore, designed with a constant circular cross section over as much of its length as practical to permit increased passenger capacity.

The entire airplane was planned for ease of maintenance. A structural floor was designed so that hatches could be cut in the belly for speedy access to equipment. In the belly compartments all equipment was grouped so there would be no interference between men working on electrical, radio, heating, and hydraulic equipment.

Finally, the basic arrangement was so planned that up-to-date power plants could be installed in the airplane to improve its utility and speed and keep it from becoming

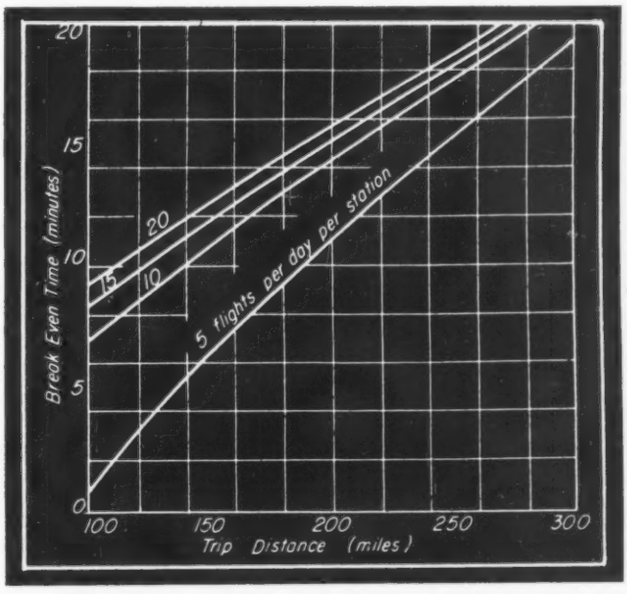


Fig. 6—Extra loading time (above five minutes) that could be allotted to a low-wing airplane for operating costs equal to those for a high-wing type

obsolete. Naturally, any airline purchasing a new airplane would like to incorporate future improvements without losing its original investment and the training of maintenance and operations personnel.

While determining the type and configuration of the final airplane to be built, numerous basic problems arose which required extensive study and research. If a new efficient airplane was to be built, all the developments made available by the war had to be incorporated. Other problems peculiar to a twin-engined transport airplane required special research programs.

A low-drag wing was required to achieve an efficient airplane with operating economy. Even though low-drag wings for high-speed airplanes were available, none fulfilled the peculiar requirements of a twin-engine transport airplane. Two major requirements were: First, low values of drag under actual flight and operational conditions although personnel are required to walk on the wing and a perfectly smooth baked enamel finish cannot be maintained as on a high-speed military airplane; second, an

airfoil whose low-drag characteristics extended through the entire speed range, rather than one with a relatively narrow low-drag bracket limiting its maximum efficiency to a narrow range of speed which may not always encompass even the desired cruise speeds. Low profile drag at high lift coefficients was needed especially during single-engine climb with one engine inoperative. A wing fulfilling these requirements was developed after extensive research, wind tunnel testing and, finally, actual flight testing.

A special wing construction was developed to achieve a smooth contour which would be non-buckling under normal flight loads and in which rubber fuel tanks could be installed. It is a so-called double-skin construction which requires few wing ribs, is light in weight, and forms a liner for the rubber Mareng fuel cells. The rubber fuel cells, incidentally, furnish a degree of safety and ease of maintenance not found in integral fuel tanks.

As mentioned before, the highest possible wing loading consistent with reasonable stall speeds and airport size is required for operating economy and fast cruising speeds. The wing flap is the means by which this can be achieved. The flaps to be used on the twin-engined transport must have: (1) A high lift coefficient for low stall speeds, (2) low drag and high lift at take-off, and (3) high lift with high drag for landing at a small airport. No currently used flap design combined all these essential features, so a new type double-slotted flap was finally developed.

To get the maximum effect of any flap characteristics on the airplane, it is necessary to extend the flap over as much of the wing as possible. On most airplanes the aileron requires 30 to 35 per cent of the wing span so here was an obvious place for improvement. The van Zelm aileron has been developed which creates as great a rolling moment as normal ailerons, with a span of only 25 per cent of the wing. This aileron is simple and requires none of the complicated mechanisms required for lateral control with full-span flap installations.

In short, the combination of the wing, flap and aileron into a unit which gives all the desired characteristics is the basic feature of the twin-engined airplane. This unit dictates the design of the airplane since airport size, cruising speed, useful load, operating cost and every important factor are dependent on it.

Among the other research programs undertaken were development of a light-weight, durable flooring and a better passenger seat design.

In this search for efficient construction, high speeds and profitable payloads, the maintenance man who keeps the airplane flying was not forgotten. Service men and engineers were sent out to the airlines to work with airline personnel and become intimately familiar with problems encountered after an airplane is put into service. The recommendations brought back by these men had a major influence on the design.

After the final airplane configuration had been determined and most of the basic problems peculiar to the airplane solved by the laboratory and research groups, detail engineering of the airplane was begun, shortly after V-E day. Like any other airplane design this workhorse of the airlines underwent changes during its detail design. Certain of the revisions were brought about by improvements in design and others by changes in the requirements of the airlines.

As expected by the manufacturer, the airlines turned their emphasis to carrying more passengers and less cargo in this type of airplane, with the majority of airlines specifying 40 instead of 30 passengers. As can be seen in *Fig.* 7, there was no change necessary in the seat spacing, passenger comfort or basic arrangement of the airplane.

From the very start of any airplane design, future developments must be studied and considered in the design. This is highly important since it is the only way to keep the original airplane from becoming obsolete.

Two modifications to the original airplane are already underway. The first new airplane, known as Martin Model 303, is intended for operations where pressureized cabins are required. Slightly higher cruising speeds are achieved by use of exhaust thrust and a smaller wing area made possible by recent elimination of the stall speed requirements. The other is the world's fastest medium-altitude twin-engined transport, employing revolutionary power plants. This airplane may be powered by turbine engines driving propellers. It will cruise at much higher speeds and offer even lower operating costs than the airplane powered with a reciprocating engine. The airplane will look very similar, having the same basic airframe.

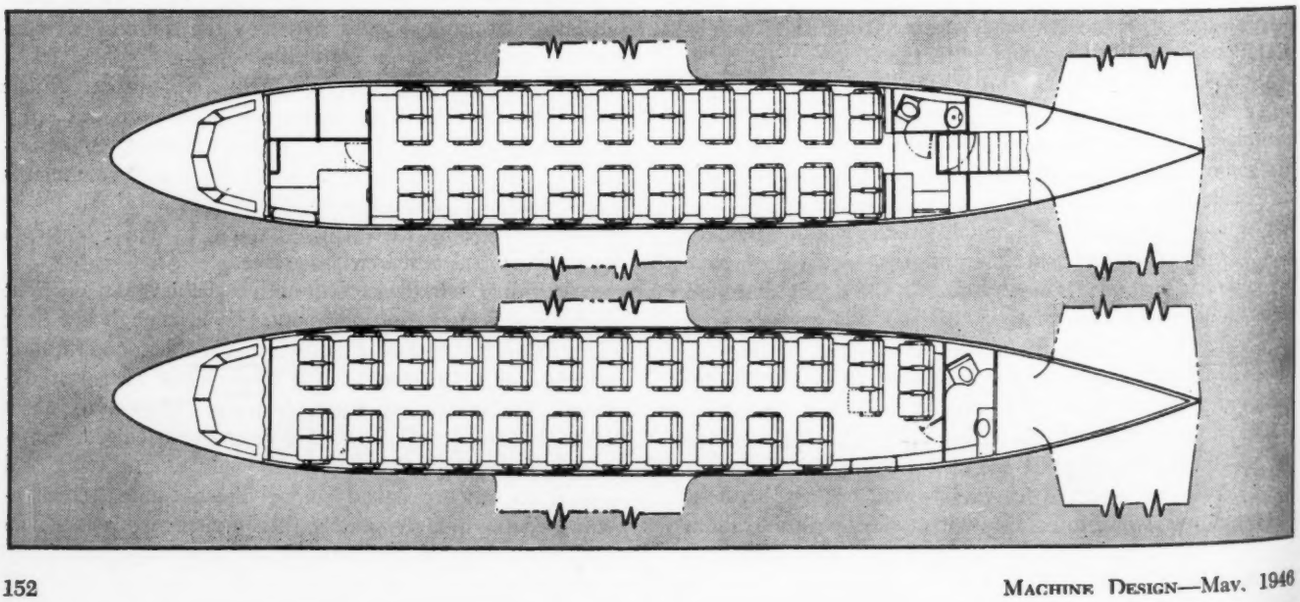


Fig. 7—Comparison of cabin seating plans for 40 and 50 passengers. Original design contemplated capacity for 30

Automatic Tension Control Improves Production

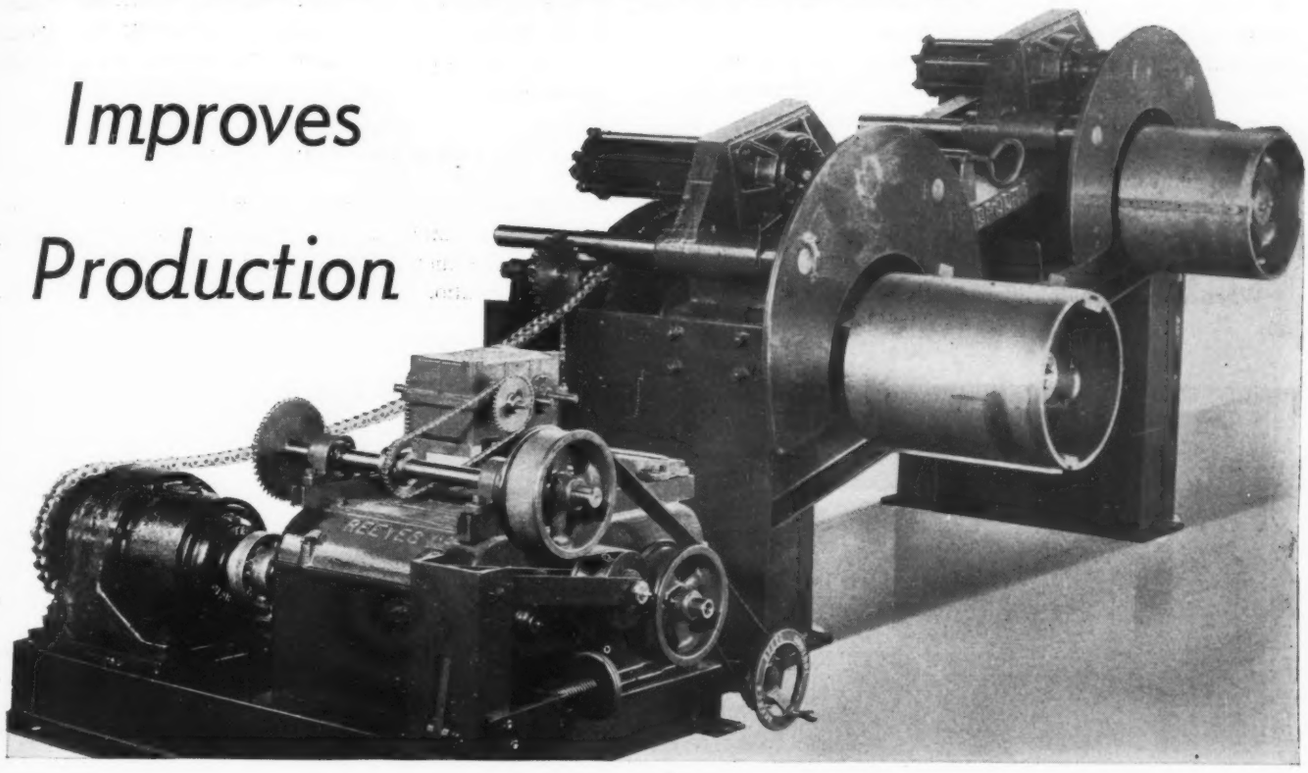


Fig. 1—Control of tension on windup reel is obtained by slip-belt type of mechanism which automatically adjusts the speed of the reel

By Joseph H. Gepfert
Division Manager
Reeves Pulley Co., Cleveland

IN processing a web of strip material it is necessary to keep the material at a constant tension, otherwise the process does not function properly. The system illustrated in *Figs.* 1 and 2 was originally developed for processing steel strip, but more recently has been used for control of the web of paper in a continuous manifold type printing press. It also is being utilized in the production of plastic film and textiles.

Success of the equipment in these widely separated fields indicate that it is a solution to a problem that has long been a source of annoyance to engineers. The basic principles of the application are set forth in this article so that they can be applied generally in solving tension control and wind-up drive problems.

Automatic control of web tension in a continuous process can be attained by using the power required to wind the material as the index for controlling the speed of the reel. Web tension, the diameter of the material on the reels and the speed of the reels determine the power required to drive the reel. Any tendency for any of these values to change is immediately reflected by a corresponding change in power required to drive the windup reel.

To maintain a constant tension in a web of material as it is wound or unwound, the speed of the reel must be continuously adjusted to compensate for the change in diameter of the material on the reel. When this is done the power required to drive the reel remains constant for any selected value of web tension.

Slip Belt Measures Power

Tension control illustrated in *Fig.* 1 consists primarily of a slip-belt mechanism for measuring the power required to drive the reel through a Reeves variable-speed transmission. The transmission is adjusted to the proper speed to maintain tension by a differential gear which measures the tendency of the "slip-belt" drive to change speed when the power required tends to change. Designation of slip belt is really a misnomer for this device, since a properly designed installation does not require any belt slippage. The measuring belt operates in the "creep" range like any other belt drive and is therefore as reliable

and maintenance free as any other belt drive.

Belt creep, or reduction in speed between the two pulleys of a belt drive, results from the tension difference between the two sides of a belt as it transmits power. The belt, in changing from the high tension on the driving or pulling side to the lower tension on the slack side, contracts slightly at the pulley and hence runs slower than the pulley. Speed difference is exactly proportional to the power transmitted up to the point that the static frictional contact is lost and the belt slips over the pulley face.

Because the speed loss is proportional to the load it is a direct indication of load change. As long as the belt is used in the creep range, no power is lost by the belt. Also, the load can be preset by selecting the tension at which the belt is operated.

When a three-shaft differential is connected with one side to the source of power, and the other side to the output side of the slip belt (in tandem with it), the third shaft indicates the difference in speed between the sides or across the belt. This shaft stands stationary when the two sides are turning at equal speeds and rotates at one half the difference between the speeds when they are not the same. By selecting the proper ratio for the connecting drives to the differential, the direction of rotation of the third shaft and the speed difference or operating load value of the slip belt can be established.

Differential Adjusts Variable-Speed Transmission

When the third shaft of the differential is connected to the speed adjusting screw of the variable-speed drive, its rotation causes a speed change in the drive which corresponds in direction and amount to that called for by the change in load, hence speed, carried by the slip belt. As long as the material is building up on the reel the tendency is to increase the load which causes the differential to operate in the direction to slow down the reel. However, any tendency in the process which might cause lower tension in the material would decrease the load and the differential would rotate in the direction to increase the speed. The latter action also takes place when the web is cut for changing reels, so the tension is held on the web automatically for all operating conditions.

The diagram in *Fig.* 2 is a schematic layout of a slip-belt automatic tension control showing the relationship of the various parts as used when paper is the web material. In addition to maintaining constant tension throughout the entire winding operation, it is possible by adjusting the weight on the tension idler, to set the tension at any time and to any value desired. When the tension in the material web is selected by adjusting the amount of weight on the tension idler, exact tension is maintained automatically.

Obviates Normal Adjustments

Since the control system is automatic, the machine operator is free to attend to other duties without fear of having a faulty product from any factor that is influenced by tension variation. This eliminates a whole series of adjustments which normally must be made. Not only does the constant-tension control reduce waste material and increase production rate of the processing machine itself but it also aids subsequent operations by providing a more uniformly wound reel.

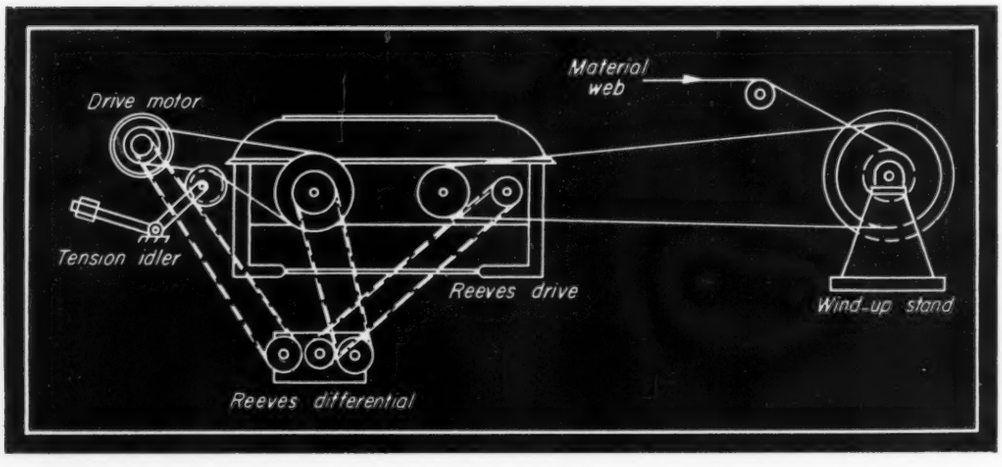


Fig. 2—Schematic diagram for automatic tension control showing the function of the slip-belt

Simplicity of the control elements and variable-speed drive allow for maintenance and repair by unskilled personnel and without special equipment. Normal care and reasonable attention to lubrication results in exceptionally long life for all of the elements.

This basic system can be used for almost any combination of web materials, web speeds and space requirements. Specific applications must be designed to fit the requirements by choosing the proper combination of parts to satisfy the specific problem.

Evils of Compulsory Licensing

ILLEGAL COMPULSORY LICENSING of patents through back-door strategy is being achieved by key government men according to the National Patent Council, a new organization of small manufacturers. John W. Anderson, president of the council, states, "Compulsory licensing would stifle inventive initiative and would remove incentive to invest money and effort in refinement, production and distribution of new products".

MACHINE *Editorial* DESIGN

Customer Acceptance Needs Emphasis!

WITH the huge backlog of orders for most types of machines that has existed since the end of war, changes in design have been relatively few and far between due to the necessity for early return to volume production. As production catches up with demand, however, it will become increasingly necessary that the refinement of existing machines and the development of entirely new machines be aimed much more intensively at meeting the requirements of the potential customer.

It is to this period that progressive machine-building companies are looking, in realization of the competitive situation that will arise as the demand for goods begins to slacken. They recognize fully that competition is apt to be more keen than ever before in view of the increased productive capacity of the nation brought about as a result of wartime developments, new and enlarged plants, and the larger number of industrial workers available.

Many machinery companies already have taken steps to insure a full measure of consumer acceptance of new designs by sending representatives to users and potential users to discuss the individual requirements that must be met. Such collaboration between maker and user often results in design improvements that might not be effected if the machine builder's engineering department were not made acquainted with the specific needs or wishes of the customer. A particularly apt instance of this procedure, incidentally, is discussed in the article in this issue dealing with the development of a new transport airplane.

In order to capitalize still further on customer cooperation, it is highly desirable that the engineer responsible for the design of a new machine be given sufficient opportunity to check the performance of pilot models under actual service conditions in various locations and different types of plants. Only by this method can the designer be expected to anticipate the peculiar needs of customers and to incorporate the largest possible number of design requirements in the final machine.

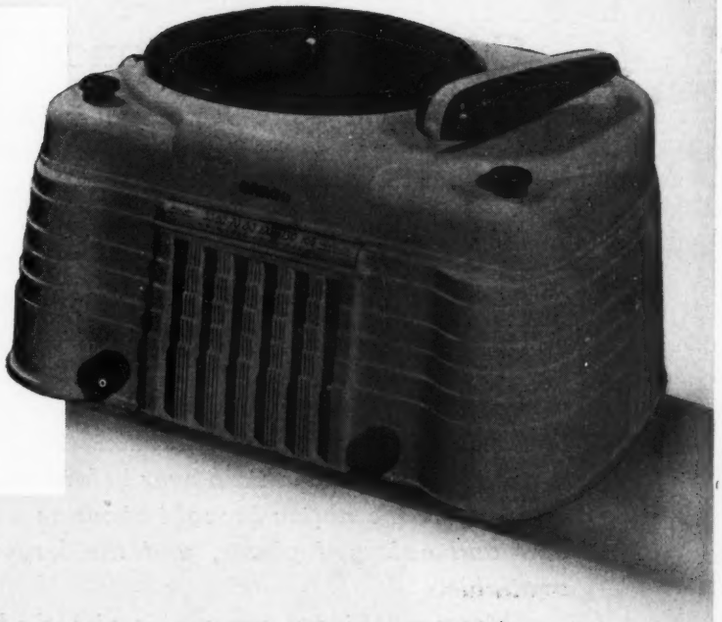
Failure to check the probability of wide customer acceptance can lead to incongruous results. The original "airflow" car—to name a well-known example—was unsuccessful from the standpoint of consumer appeal and was changed radically in later models. In that instance it was evident that the new car was far ahead of its time and that insufficient attention seemingly had been paid to the requirements of the market as they then existed. Timeliness in design is an all-important factor; its accuracy can be gaged only by painstaking study of the market.

L. E. Jermy

Outstanding Designs

Radio-Phonograph

Housing of this radio-phonograph is one-piece compression-molded plastic, with ivory finish. Its superheterodyne circuit utilizes five multipurpose tubes for high fidelity broadcast reception and widest tonal range for 10 or 12-inch records. For a-c operation on 105-125 volts, 50-60 cycles, this combination instrument employs a self-starting motor, crystal pick-up, phono-radio switch and volume control in addition to the tone and tuning controls. Manufacturer: Garod Radio Corp., Brooklyn, N. Y.

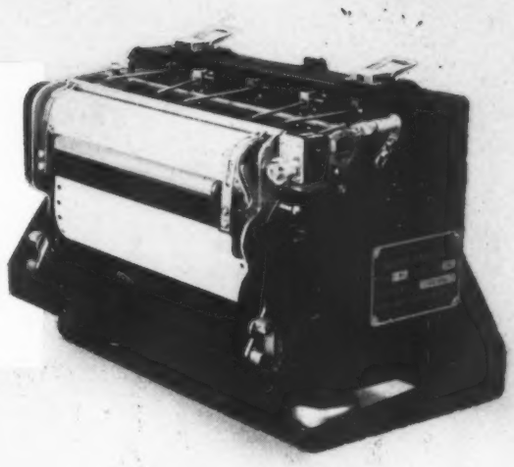


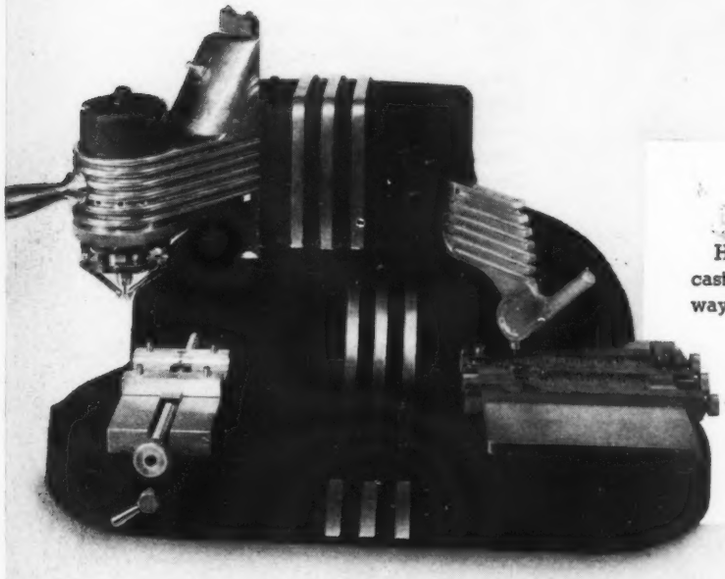
Clothes Washer

In its four-cycle process of washing, rinsing, drying and fluffing, this automatic washer imparts to its basket a combination bouncing and spinning action. Clothes are inserted through the top opening into the perforated aluminum and magnesium basket which is equipped with a series of smooth, rounded baffles. Jets at bottom of basket spurt soapy water during washing cycle. An adjustable thermostat can be set to hold wash water at 100 or 130 F. Rinse water is automatically maintained at 100 F. Automatic cut-off switch on lid stops all machine action when lid is lifted. Interior lighting makes possible viewing of the process through the transparent cover. Manufacturer: Apex Electrical Mfg. Co., Cleveland.

Flight Analyzer

In this machine, flight functions of an airplane are recorded on a traveling chart. Recording is done without ink, the lines being scribed by electrically charged styluses on Teledeltos paper. A special switch turns on the analyzer only when the plane is moving at over 75 miles per hour. Manufacturer: Hathaway Instrument Co., Denver.



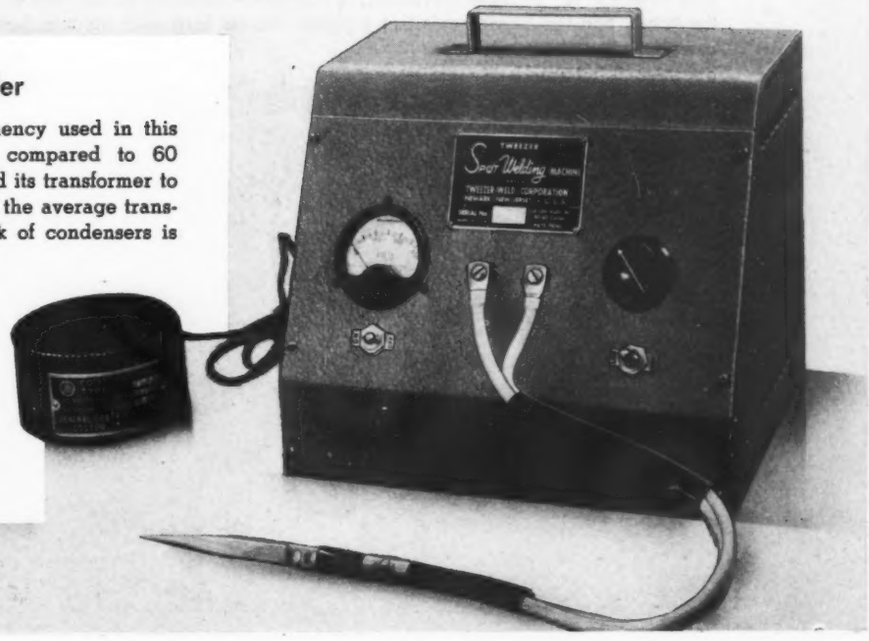


Engraving Machine

Housings and casings of this engraver are cast aluminum and its hardened and ground ways ride on hardened precision steel balls. Motor uses preloaded ball bearings and operates at approximately 6000 rpm under load. Spindle housing has a micrometer depth-controlled follower. Manufacturer: Auto Engraver Co., New York City.

Spot Welder

Because of the high frequency used in this "tweezer" spot welder, as compared to 60 cycles, it was possible to build its transformer to about one-tenth the weight of the average transformer. In operation, a bank of condensers is charged to d-c voltage and discharged through a relay and the transformer by operation of the foot switch. Machine is encased in gray, wrinkle finish, steel carrying case. Manufacturer: Tweezer-Weld Corp., Newark, N. J.



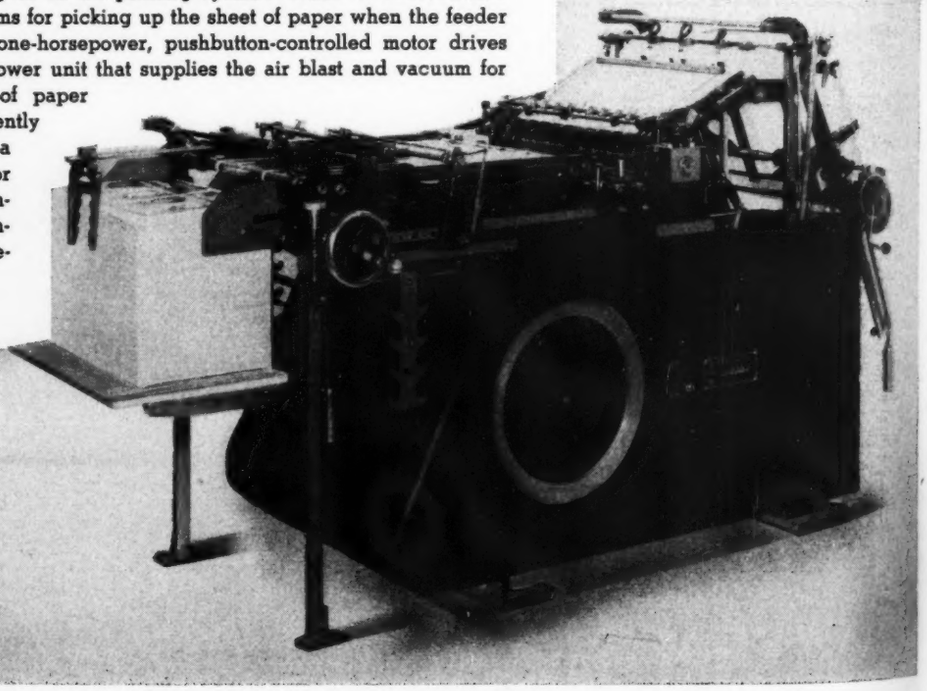
Concrete Mixer

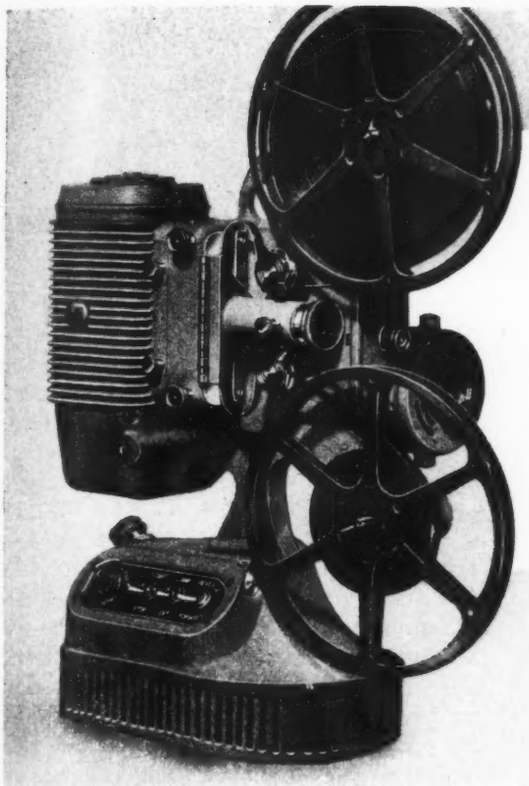
For ease in maneuvering, towing and handling, this concrete mixer has been designed with a low center of gravity. All parts in contact with material are abrasion-resistant steel and all framing subassemblies are welded construction. The all-steel welded drum has cold-pressed die-formed heads with integral tracks. It revolves on cast semisteel rollers mounted on stationary shafts with two Timken bearings in each roller. Lubrication fittings are at ends of the shaft.

Steel-forged sprocket segments are bolted to the drum, and power is transmitted from a countershaft by chain because the chain absorbs the shocks encountered during mixing. Double-strand roller chain is used between the engine power-takeoff shaft and the countershaft. Engine clutch and power-loader clutch are multiple-disk type. Manufacturer: Chain Belt Co., Milwaukee.

Automatic Cylinder Press

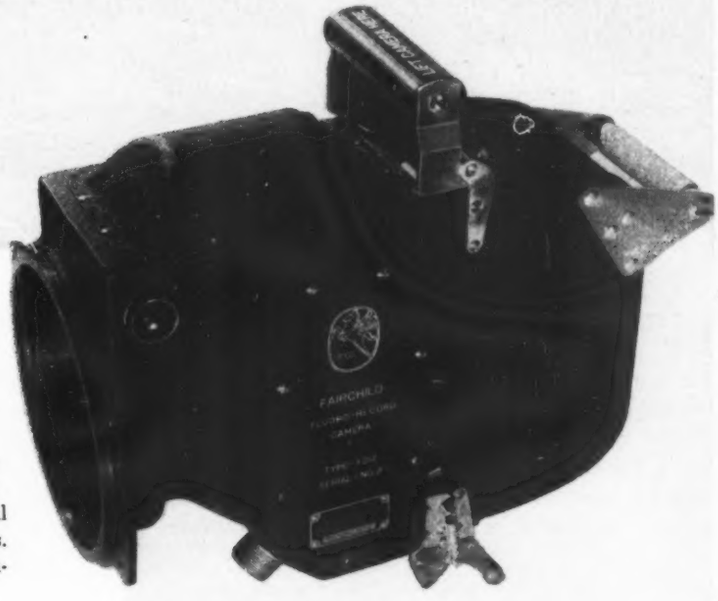
Cast iron base of this automatic cylinder press is so constructed that the bed, which has full-length scraped ways, has three-point support. Crank drive for table movement is from large bull gear. A rack on the table meshes with a gear on the printing cylinder which in turn actuates the various mechanisms for picking up the sheet of paper when the feeder arm delivers it. A one-horsepower, pushbutton-controlled motor drives the press, and the blower unit that supplies the air blast and vacuum for carrying the sheet of paper operates independently of the press with a 1/3-horsepower motor driving a pump. Manufacturer: Banthin Engineering Co., Bridgeport, Conn.





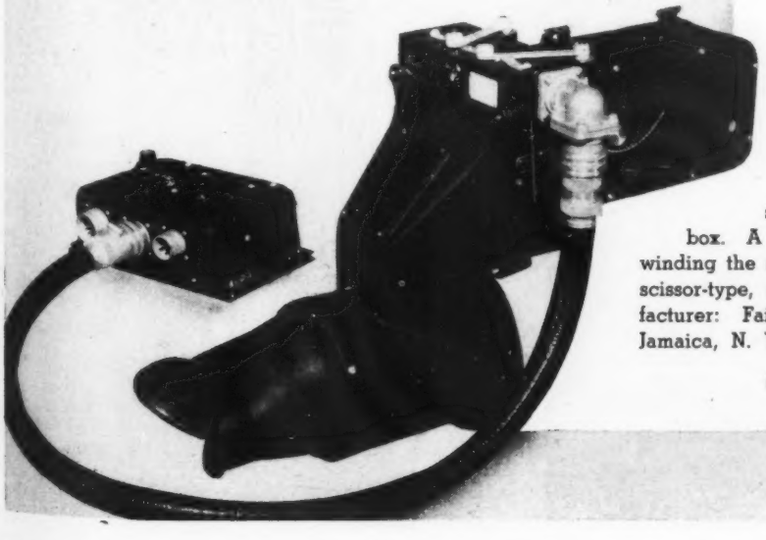
Movie Projector

Important controls of this new movie projector are grouped within a panel on one side of the machine for ease of operation. Film gate is hinged to facilitate threading and cleaning. Side and back of lamp housing, cast as one piece, are removable by loosening four knurled screws. Smooth action is achieved by utilizing machine-cut gears, oilless bearings, a friction drive, and automatic clutches. Film motion is accomplished by a gear and chain drive to a double-claw movement. External housing parts are die-cast aluminum and zinc with blue-gray wrinkle finish. Manufacturer: De Jur-Amsco Corp., Long Island City, N. Y.



X-Ray Camera

This camera comprises a body with lens, motor, electrical connections and drive mechanism, and a removable film magazine containing an electrical interlock circuit and the remainder of the drive mechanism. Lens used is designated the "Kodak-X-ray Ektar", aperture f-1.5, focal length, 111 mm and consists of seven elements. Manufacturer: Fairchild Camera & Instrument Corp., Jamaica, N. Y.



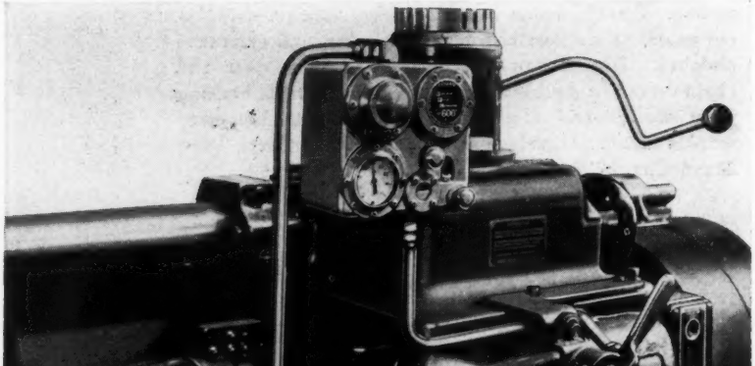
Radar Camera

Designed to make photographic records of the luminous images appearing on the cathode-ray tube screen of a radar set, this equipment comprises a camera proper, a magazine, a beam-splitter, an adapter casting, and a control box. A solenoid-actuated mechanism is used for winding the spring that moves the 35-mm film and a scissor-type, or rotary, shutter is employed. Manufacturer: Fairchild Camera & Instrument Corp., Jamaica, N. Y.

(New machines listed on Page 192)

Applications

of Engineering Parts, Materials and Processes

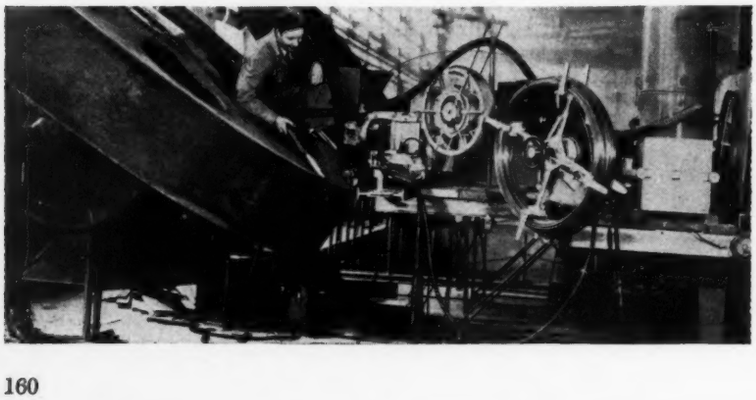


Flexible Suspension

HEAD suspension of the Barnes 924 vertical drilling, boring, and facing machine, right, is accomplished by a counterweighting system using multiple-strand roller chain as the flexible connecting medium. Having an adjustable counterweight, various heads within a 3100-pound weight range may be balanced. Very slow feeds, achieved by means of a hydraulic system, the piston of which may be seen below the head saddle, are made feasible through the relief in loading effected by the suspension system.

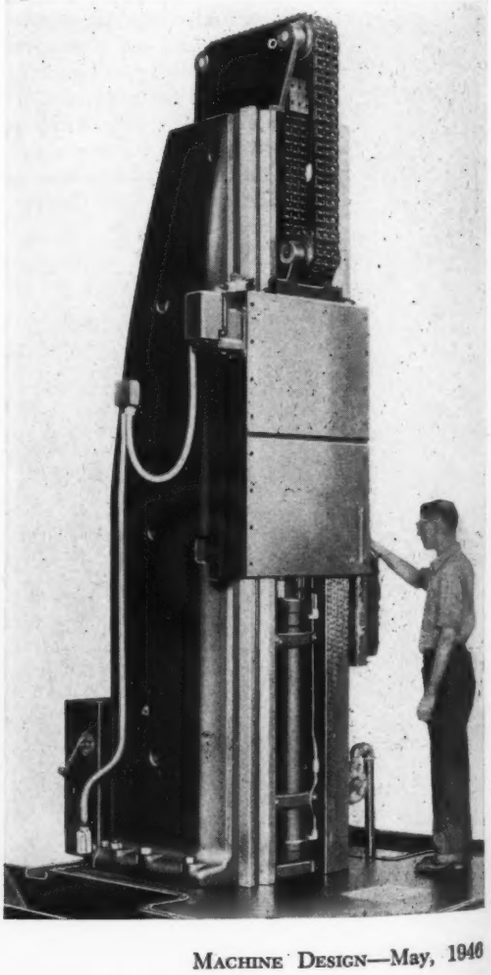
Welded Gears

AUTOMATIC welding in one pass of 12-foot diameter marine reduction gears, greatly speeding production, has been achieved through the use of the Unionmelt process. A cantilever-supported stationary welding machine, below, supplies welding rod to the work from 150 lb coils, producing a uniform fillet weld 1⅛-inches wide, and 36 feet in length. With the gear mounted on a driven, revolving stand, the welding process proceeds at high speed under a granular flux, thereby dissipating little heat to the area adjacent to the weld.



Simplifies Air Control

COMBINING pressure regulator, filter, and lubricator, all installed within a casting measuring 6 inches wide, 6 inches high, and 4¾-inches deep, the Logan "R-F-L" unit installed on a turret lathe, left, serves to supply air-operated equipment with clean, dry, "mist-lubricated" air at an adjustable pressure. The unit eliminates the old "Christmas tree" assembly of reducing valve, pressure gage, filter, and lubricating device, and the attendant maze of piping and fittings. Not only is the friction loss in the pipe minimized, but a cleaner, more compact installation results. The complete control is suitable for incorporation in the design of new machines, or may be installed on equipment already in service.



Lubricant Selection Simplified

By James G. O'Neill
Staff Engineer
Sinclair Refining Co.
New York

FOR the purpose of simplifying the selection of lubricants, general lubrication charts are presented in this Data Sheet. Various types of mechanical equipment are covered including engines, turbines, motors, pumps, machine tools, gear reducers, couplings, drives, and shafting, as well as specialized steel mill requirements.

Lubrication has played a principal role in maintaining maximum production. Lubrication engineers realizing the difficulties of replacing and repairing heavy duty machinery, designed lubricants to partially correct the mechanical conditions and extend service life. Certain specialized lubricants were adapted to service wear, leakage, shock loads, and additional bearing pressures.

As is the case with many new products, the field of their application has broadened. These specialized lubricants are often used on machinery other than that for which they were originally intended. As time goes on, their use is spreading to such an extent that reduction in the number of plant lubricants through broader and more general application appears warranted. However, the very nature of these lubricants limits them to certain fields of application. Thus, it is still necessary to use con-

General Industrial Lubrication Chart

Shafting

	S.U. Viscosity at 100° F	Oil Class
Bearings, Oil Lubricated	200-600	1, 2, 3 and 4

NOTE: Special leak preventing oils of 300 to 1600 S.U. Vis. at 100° are sometimes found advantageous.

Plain Bearings, Grease Lubricated	Calcium soap grease in N.L.G.I. 2 range with oil of 250-800 S.U. Viscosity at 100° F.
Anti-Friction Bearings, Grease Lubricated	Soda soap grease (meeting A.B.E.C. qualification for general purpose anti-friction bearing grease) in N.L.G.I. number 1 and 2 ranges.

Flexible Couplings

	GEAR, FLOATING KEY & FLEXIBLE PIN TYPES, OIL LUBRICATED		ROLLER, SILENT CHAIN & JAW TYPES — Oil		GREASE	OPEN
Surrounding Temperature	S.U. Viscosity at 100° F	Oil Class	S.U. Viscosity at 100° F	Oil Class	Soda Soap with oil 80-150 S.U. Vis. at 210° F	(Without Casing)
0° F- 70° F	1500-2500	2, 3 and 4	350- 600	1, 2, 3 and 4	N.L.G.I. No. 2	Special care required. Proper lubricant depends on the operating speed and the surrounding conditions.
70° F-120° F	2000-5000	2, 3 and 4	490- 700	2, 3 and 4	N.L.G.I. No. 2	
120° F-180° F	2000-5000	2, 3 and 4	700-1000	2, 3 and 4	N.L.G.I. No. 3	

Electric Motors and Motor Generators

	BEARINGS, OIL LUBRICATED		BALL & ROLLER BEARINGS, GREASE LUBRICATED
Surrounding Temperature	S.U. Viscosity at 100° F	Oil Class	Soda Soap Grease (meeting A.B.E.C. qualification for general purpose anti-friction bearing grease)
0° F- 70° F	140-275	1 and 2	N.L.G.I. No. 1
70° F-120° F	140-275	1 and 2	N.L.G.I. No. 1 and 2
120° F-180° F	250-325	1 and 2	N.L.G.I. No. 2

Chain Belt Drives

SLOW SPEED DRIVES 0° F-180° F	MEDIUM SPEED DRIVES 0° F-180° F	HIGH SPEED DRIVES 0° F-120° F	HIGH SPEED DRIVES 120° F-180° F
S.U. Vis. at 100° F 300-600	S.U. Vis. at 100° F 200-600	S.U. Vis. at 100° F 140-275	S.U. Vis. at 100° F 225-325
Oil Class .....1, 2, 3 and 4	Oil Class .....1, 2, 3 and 4	Oil Class .....1, 2, 3 and 4	Oil Class .....1, 2, 3 and 4

Engineering Data Sheet

Gear Reducers

("Tentative AGMA (American Gear Manufacturers Association) Standard Specification—Gears 250.01")

Table I—Viscosity Range for Various Agma Lubricants

AGMA Lubricant No.	Viscosity Range at 100° F	S.U. Vis. Seconds at 210° F
1	180 to 240	.....
2	280 to 360	.....
3	490 to 700	.....
4	700 to 1,000	.....
5	.....	80 to 105
6	.....	105 to 125
7	.....	125 to 150
7 Comp.*	.....	125 to 150
8	.....	150 to 190
8 Comp.*	.....	150 to 190
9	.....	350 to 550
10	.....	900 to 1,200
11	.....	1,800 to 2,500

* The oils marked "Comp." are those compounded with 3 to 10 per cent of acidless tallow or other suitable animal fat.

Oils No. 9, 10 and 11 should be of a tacky adhesive nature so that they will adhere well to the tooth surfaces.

Table II—Recommended Lubricants for Enclosed Units of All Types Except Worm Gears

Type and Size of Units	Ambient Temperature, deg. F: 0-40 use AGMA No.	41-100 use AGMA No.	101-150 use AGMA No.
Parallel Shaft Units			
Low-speed, Centers up to 20 in.	2	4	5
Low-speed, Centers over 20 in.	3	5	6
Planetary Gear Units			
O.D. of housing up to 16 in.	2	3	4
O.D. of housing over 16 in.	3	4	5
Gearmotors			
All sizes	2	4	5
Spiral or Straight Bevel Gear Units			
Cone distance up to 12 in.	2	4	5
Cone distance over 12 in.	3	5	6
High Speed Units			
All sizes	1	2	3

NOTE: Speed range = 600-1800 RPM — High Speed = above 1800 RPM — Gears heat treated after cutting use next higher AGMA No

Table III—Recommended Lubricants for Enclosed Worm Gear Units Only

Service and Worm Speed	Ambient Temperature, deg. F: 0-40 use AGMA No.	41-90 use AGMA No.	91-120 use AGMA No.
Intermittent operation — all worm speeds	5	5	7 Comp.
Continuous operation—worm speeds below 600 rpm	7 Comp. † Diluted	8 Comp.	8 Comp.
Continuous operation—worm speeds 600 rpm and over	7 Comp. † Diluted	7 Comp.	8 Comp.

* Where period of operation is insufficient to produce any appreciable rise in oil bath temperature.

† Diluted AGMA No. 7 comp. oil should be diluted with a lighter oil, preferably not exceeding 500 sec. viscosity at 100° F, until the desired fluidity is obtained. The lubricant used for dilution should be of the same basic crude as that of the recommended oil. The lubricant supplier should be consulted if there is any doubt.

Table IV—Recommended Lubricants for Open Gearing of All Types Except Worm Gears

Method	Ambient Temperature, deg. F: 0-40 use AGMA No.	41-100 use AGMA No.	101-150 use AGMA No.
Slush Pans	3	5	6
Hot—by brush or paddle*	10	10	11
Cold—by brush or paddle*	6	8	10
Hand-oiled	4	6	8

*Also for wire ropes and cables.

Table V—Recommended Lubricants for Open Worm Gears Only

Method	Ambient Temperature, deg. F: 0-40 use AGMA No.	41-100 use AGMA No.	101-150 use AGMA No.
Cold or hot—by brush or paddle	6	8	..

NOTE: Ambient = surrounding air temperature.

Steam Turbines and Generators

	S.U. Viscosity at 100° F	Oil Class
Direct Connected Turbo-Generator Sets (circulating system)	140-200	1 and 2
Geared Turbines (circulating system)	250-325	1 and 2
Turbines, Ring Oiled Bearings, With Cooling System (not oil circulating system)	250-325	1 and 2
Turbines, Ring Oiled Bearings, Without Cooling System (not oil circulating system)	300-600	1 and 2

Steam Engines and Steam Pumps

	Steam Temperature	
Cylinders (not circulating system)	Below 370° F. (to 160 lbs. pressure)	115-150 S.U. Vis. at 210° F with 6-10% acidless tallow, etc.
	370° F.-500° F.	140-160 S.U. Vis. at 210° F with 4-8% acidless tallow, etc.
	500° F.-700° F.	150-190 S.U. Vis. at 210° F with 3-6% acidless tallow, etc.
	Above 700° F.	190-225 S.U. Vis. at 210° F (straight mineral)

	S.U. Viscosity at 100° F	Oil Class
External Lubrication, Oil	200-600	1, 2, 3 and 4
External Lubrication, Grease	Calcium soap grease in N.L.G.I. (National Lubricating Grease Institute) 2 and 3 ranges with oil of 250-800 S.U. Viscosity at 100° F	
Circulating Systems	175-600	1, 2 and 3

Engineering Data Sheet

Stokers

Gear Reducers and Cables	(see "Gear Reducers")		
Hydraulic Pumps	(see "Hydraulic Pumps and Their Systems")		
		S.U. Viscosity at 100° F	Oil Class
Roller and Silent Chains and General Lubrication		175-600	1, 2, 3 and 4
Plain Bearings, Grease	Calcium soap grease in N.L.G.I. 2 and 3 ranges with oil of 250-800 S.U. Viscosity at 100° F.		
Anti-Friction Bearings, Grease	Soda soap grease (meeting A.B.E.C. qualifications for general purpose anti-friction bearing grease) in N.L.G.I. number 1 and 2 ranges.		

Compressors

	Equivalent SAE Grade	S.U. Viscosity at 100° F	Oil Class
Air and Gas Compressors (follow manufacturers instructions)	10	180-240	1, 2 and 3
	20	280-360	1, 2 and 3
	30	490-700	1, 2 and 3
Refrigerating Compressors (follow manufacturers instructions)	..	135-525	Low pour point moisture free oils. Not shown in this chart.

Diesel Engines

	Equivalent SAE Grade	S.U. Viscosity at 100° F	Oil Class
(Follow manufacturers recommendations)	10	180-240	2 and 3 and
	20	280-360	detergent
	30	490-700	detergent
	40	700-1000	detergent
	50	80-105 (at 210° F)	

Machine Tools

Hydraulic Systems	(see "Hydraulic Pumps and Their Systems")		
		S.U. Viscosity at 100° F	Oil Class
Circulating Systems		140-200	1 and 2
Automatic Lubricating Systems and General Lubrication of Slides, Ways, Bearings, etc., Oil		200-600	1, 2, 3 and 4
Anti-Friction and Plain Bearings, Grease	Soda soap grease (meeting A.B.E.C. qualifications for general purpose anti-friction bearing grease) in N.L.G.I. number 1 and 2 ranges.		
Gear Boxes	(see "Gear Reducers")		
Spindles, Separately Lubricated	(Follow manufacturer's instructions.)	45-125	Highly refined light viscosity oils.
Electric Motors	(see "Electric Motors and Motor Generators")		

Hydraulic Pumps & Their Systems

Surrounding Temperature	VANE TYPE with Flow Control Valves S.U. Viscosity at 100° F	Oil Class	VANE TYPE without Flow Control Valves S.U. Viscosity at 100° F	Oil Class	GEAR & PISTON TYPE S.U. Viscosity at 100° F.	Oil Class
0° F- 70° F	140-200	1 and 2	225-275	1 and 2	140-200	1 and 2
70° F-120° F	140-200	1 and 2	225-275	1 and 2	250-325	1 and 2
120° F-180° F	140-200	1 and 2	225-275	1 and 2	300-525	1 and 2

NOTE: For below 25° F. and above 150° F on the Vane Type Pumps special instructions are issued by the manufacturer.

Description of Oil

Class 1—Specially refined, additive, turbine-type lubricating oils. Meet and exceed requirements of makers of steam turbines, hydraulic pumps, etc.

Class 2—Highly refined, non-additive, turbine-type lubricating oils. Enviable reputation for indicated uses.

Class 3—Highly refined, non-additive, coastal lubricating oils. Very outstanding for indicated uses.

Class 4—General-purpose oils: Conventionally refined, non-additive, general purpose oils. Excellent for General lubrication but not adapted to specialized operating conditions of critical units.

NOTE: A properly selected oil may service a wide variety of mechanical units. By proper selection, inventory can be lessened and savings in storage, errors in application, etc. achieved.

ventional lubricants for services which cannot be satisfactorily handled by new-type products.

Lubrication requirements of a wide variety of mechanical units used by the metalworking industry have been established, as set forth in the charts. Specialized lubricants should not be used except under supervision or with the advice of a competent lubrication engineer.

It is important to note that the specialized lubricants are each adapted to specific conditions. Following are the more commonly accepted applications and the general makeup of each of these:

Leak-preventing oils are manufactured in five grades. Their makeup consists of black oils plus a synthetic additive which gives the oils adhesive qualities and thus reduces leakage. The generally encountered Saybolt Universal viscosities at 100 F are as follows: 500-700, 1000-1200, 1500-1800, 2000-2200 and 2500-2700.

These lubricants are not intended for enclosed circulating systems. Their principal application is on oil-lubricated bearings where leakage must be counteracted and where temperatures are not extreme. Examples are oil-lubricated shaft bearings and miscellaneous heavy-duty bearings, such as car wheel journals, general mill bearings, and conveyor system bearings. The selection of the proper

leak-preventing lubricant is made on the basis of substituting the equivalent viscosity product for the straight mineral oil in use.

Extreme pressure oils are designed primarily for use in worm gear reducers or reducers subjected to overloads and shock loading. They are also sometimes found advantageous in oil pressure lubricating systems of the Bowser, DeLaval, etc., types, for lubricating table rolls, pinions, and drive gear sets under heavy-duty service. The selection of the proper oil is made on the basis of reference to the straight mineral oil ordinarily used for the purpose. These oils should not be used on worm gear sets if the manufacturer specifies an oil containing tallow compound.

Extreme pressure oils are manufactured in ten grades. They consist of black oils and a lead compounded additive. Properties of film strength, shock load absorption, and leakage prevention are incorporated in their make up. The viscosities at 100° F and 210° F are as follows:

Oil	Saybolt Universal Viscosity at 100° F	Saybolt Universal Viscosity at 210° F	Substitute for AGMA No.
No. 1	300	50	2
No. 2	750	70	4
No. 3	1280	89	5
No. 4	1890	110	6
No. 5	2400	125	7
No. 6	3400	150	8
No. 7	....	210	
No. 8	....	430	9
No. 9	....	520	9
No. 10	....	950	10

Calcium soap EP greases are made in three N.L.G.I. (National Lubricating Grease Institute) grades, classification Numbers 0, 1, and 2. They incorporate calcium soap and a dark-colored, high-viscosity oil. In addition, they have good film strength properties. In general, their application replaces, for steel mill use, the more conventional cup and bearing greases.

Field of application of these greases is the lubrication of bearings under normal temperatures. Miscellaneous heavy-duty, grease-lubricated bearings such as car wheel journals, general mill bearings, and conveyor system bearings, can often use them to advantage. These lubricants are also employed in grease pressure lubricating systems, under normal temperature operation, for table rolls, heavy-duty roll-neck bearings, pinions, drive gears, etc., using Trabon, Bosch, B-K, Farval, Alemite, Lincoln, etc., systems.

Present practice in many plants tends to the use of calcium soap EP greases, both as plain bearing greases and general purpose antifriction bearing greases. This reduces the number of greases in the inventory. However, when considering the use of any calcium soap grease in antifriction bearings, the advice of the plant lubrication engineer should be followed. High-temperature service and speeds (above 175 F and 2000 rpm) prevailing on such bearings requires the use of conventional soda soap greases.

Soda soap graphite grease is made using a high viscosity oil and about 8 per cent graphite. It is of the N.L.G.I. No. 2 consistency and is designed for difficult lubrication problems, such as hydraulic plunger mechanisms and plain bearings subjected to excessive water wash and other drastic conditions. It is also recommended for lubricating sintering plant, pallet conveyor wheel bearings.

Iron and Steel Mill Specialized Lubrication Requirements

Shafting	
Bearings, Oil	OIL: Leak Preventing Oils
Bearings, Grease	GREASE: Normal Temperature Service— Calcium Soap EP Greases Nos. 0, 1, 2 High Temperature Service— Heavy Duty Soda Soap Greases Nos. 0, 1, 2, 3
Miscellaneous Heavy Duty Bearings	
Car Wheel Journals General Mill Bearings Conveyer System Bearings	OIL: Leak Preventing Oils
Car Wheel Journals General Mill Bearings Conveyor System Bearings	GREASE: Normal Temperature— Calcium Soap Greases Nos. 0, 1, 2 High Temperature— Heavy Duty Soda Soap Greases Nos. 0, 1, 2, 3
Hydraulic Plunger Mechanisms Plain Bearings subjected to excessive water wash, etc. Sintering Plant, Pallet Conveyor, Wheel Bearings	GREASE: Soda Soap Graphite Grease

Pressure Lubrication Systems	
Table Rolls, Pinions, Drive Gear Sets, etc. (Bowser, DeLaval, etc.)	OIL: Normal Service— General Purpose Oils Heavy Duty Service— Extreme Pressure Oils
Table Rolls, Pinions, Drive Gears, etc. (Trabon, Bosch, B-K, Farval, Alemite, Lincoln, etc.)	GREASE: Normal Temperature Service— Calcium Soap EP Grease Nos. 0, 1, 2 High Temperature Service— Heavy Duty Soda Soap
Heavy Duty Roll Neck Bearings	GREASE: Calcium Soap EP Greases

Gear Reducers

The selection of the extreme pressure oils is made by substituting for the applications shown in the "General Industrial Lubrication Chart" the substitute AGMA Number of extreme pressure oil.

Heavy-duty, slow-speed units in oil type cases are often lubricated to advantage by use of extreme pressure oils Nos. 9 and 10. These oils have adhesive qualities superior to the conventional oils normally recommended.

LONG LIFE • LOW COST • QUIET • SMOOTH OPERATION PRECISION • EASY
RESISTANCE TO POUNDING • LOW FRICTION • HIGH LOAD CARRYING CAPAC
RESISTANCE TO SHOCK • WATER PROOF ION • ECO
UNIFORM • ATION PREC
ASSEMBLY OAD CARR
TOUGH • CORROSI
ECONOMICA OOTH OP
LOW COST • IBLY • R
POUNDING TOUGH
SHOCK • W AL • UN
ONG LIFE • EAS
RESISTANCE TO APACIT
STANCE TO SH CONON
• UNIFORM • PRECI
LIFE • LOW CO SSEM
ANCE TO POUND Y •
ANCE TO SHOCK OMIC
RM IFE ON
ANCI ING
H • CE TO CO ON •
OM ERATIO
OW COST • PR EASY A
RESI PO FRICTION • HIGH LO CARRY IG IT
RESIS PROO STANCE SION
UNI COST QUIE OPERA RECI
ASSEM • RESI ANCE O POUNDING • L FRICTION • HIGH LOAD CARRY

YOU GET
Every Good Bearing Quality
in JOHNSON BRONZE
SLEEVE TYPE BEARINGS

There are many factors that govern a wise choice of bearings for a specific application. It might be purely economic such as low initial cost . . . quick assembly . . . easy replacement. It might be toughness . . . the ability to withstand terrific shock and impact . . . to carry heavy loads without danger of failure. Or it might be quiet operation . . . smooth, silent performance. It may be precision . . . fine workmanship adherence to close tolerances. In some cases it is all of these and more.

The most important factor, however, is the name and reputation of the bearing manufacturer.

In order to produce bearings that will deliver the greatest performance . . . for the longest period of time . . . at the lowest cost . . . requires considerable experience, complete facilities, up-to-date equipment, skilled employees. This is the kind of Bearing Service Johnson Bronze offers manufacturers of all types of equipment.

JOHNSON BRONZE COMPANY
525 SOUTH MILL STREET NEW CASTLE, PA.

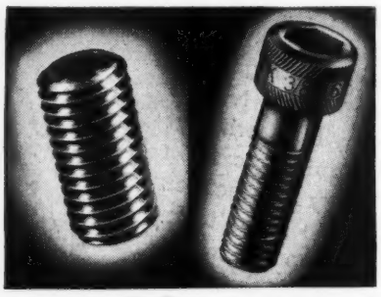
JOHNSON BRONZE
SLEEVE BEARING HEADQUARTERS

BRANCHES IN 18 INDUSTRIAL CENTERS

NEW PARTS AND MATERIALS

Socket Screws Improved

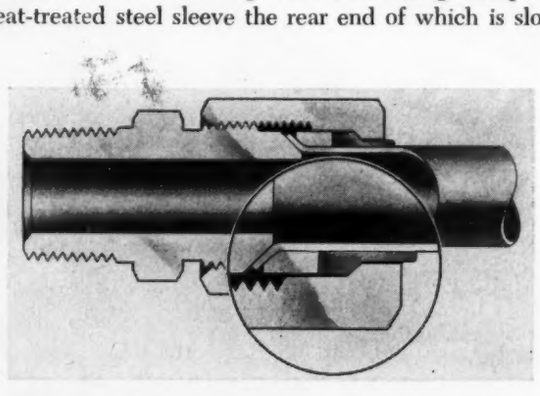
THREE IMPROVEMENTS to socket screws have been announced by Parker-Kalon Corp., 200 Varick St., New York 14. These consist of adapting centerless thread grinding to the company's socket set screws, and adding the size mark and gear grip to the socket head cap screws.



Thread grinding provides a smooth finish, with no nicks or imperfections. Marking sizes on socket head cap screws makes apparent the correct size of each screw at a glance. The new gear grip feature of the cap screws prevents slipping and fumbling and speeds assembly.

High-Strength Tube Fittings

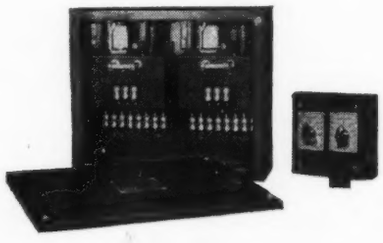
TUBE fittings featuring unusual construction have been announced by the Flodar Corp., 331 Frankfort Ave., Cleveland. Known as "Grip Tube", the fitting incorporates a heat-treated steel sleeve the rear end of which is slotted



to form spring fingers. Tightening of the nut forces the forward end of the sleeve against the tube flare, effectively sealing the fitting. Further slight tightening of the nut contracts the segmental fingers of the sleeve around the tube, thus gripping the tube uniformly. Vibration is absorbed owing to the cantilever spring support. It is claimed that the heat-treated sleeve and the flare seal joint will withstand pressures in excess of tube ratings; also that the tube will not expand at the flare nor the sleeve expand in tightening and thus cause jamming within the nut. The tube fittings, which are available in ¼ to 2-inch tube sizes, may be re-used repeatedly.

Repeat-Cycle Timer

DESIGNED FOR machinery and process control, a new repeat-cycle timer is recommended for applications requiring two adjustable timing periods to run in a continuous cycle. The cycle is initiated by either momentary or sustained contacts, and provision for automatic recycling is provided. Each timing period is adjustable from 1/20th second to two minutes. Control is accomplished through specially-designed snap-action relays. This timer, known as Type 2T15U and produced by Photoswitch Inc., 77 Broadway, Cambridge 42, Mass., is universal for six maximum time ranges from 1.5 seconds



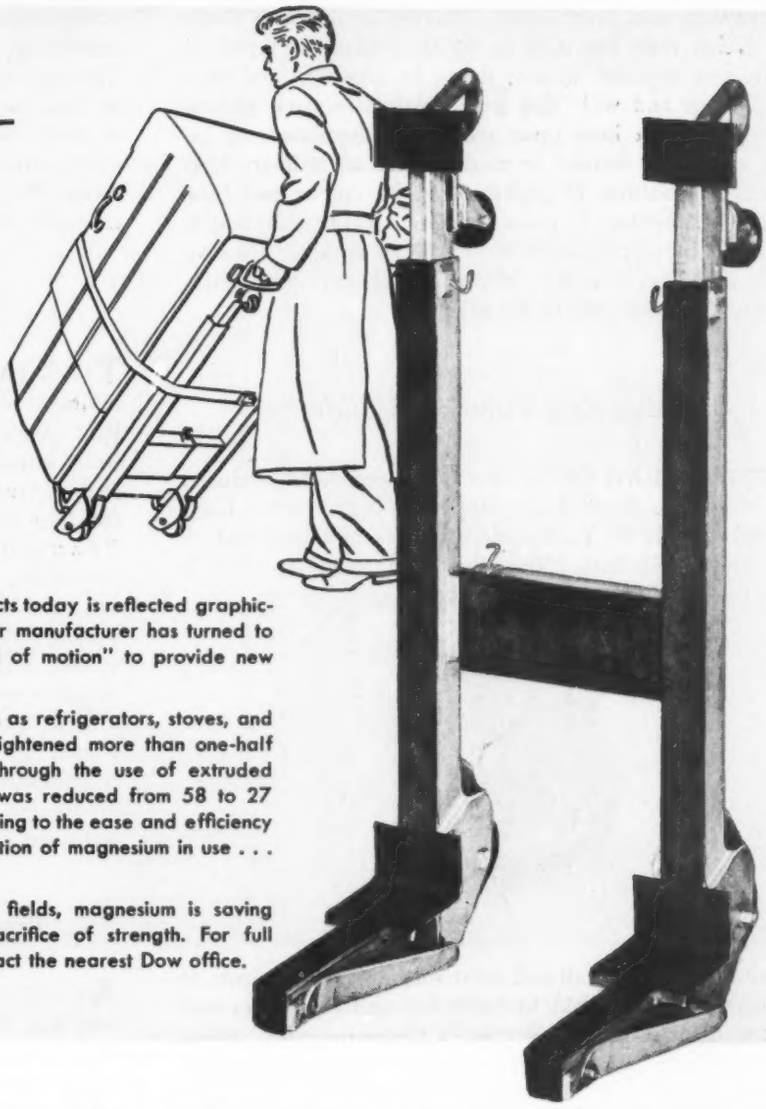
to two minutes for each period of the cycle. Each range is represented by a timing element which is snapped into a clip on the front of the control. Timer may be then set for any intervals up to these maximums by dials, located either on the timer itself or in a small housing at a more convenient location. Supplied for operation at 115 volts alternating or direct current, the timer has output connections of a single-pole double-throw switch for normally-open and normally-closed operation. Contacts are 1000 watts at 115 to 600 volts alternating current, and 500 watts at 115 to 220 volts direct current.

Improved Gasketing

KNOWN AS Chrome Lock and produced by Products Research Co., 634 South Western Ave., Dept. A109, Los Angeles 5, a new gasketing is flame resisting, fuel resisting, rust inhibiting, antiwicking, nonoxidizing, compressible, and cushioning. It also assures longer life, ease

Modern weight lifter—

lightened for new efficiency!



The "lift" magnesium is giving many products today is reflected graphically in the Keen hand truck. Here another manufacturer has turned to the lightness and strength of "the metal of motion" to provide new efficiency of operation.

Especially designed for such heavy loads as refrigerators, stoves, and similar appliances, this hand truck was lightened more than one-half by the Keen Manufacturing Company through the use of extruded shape and sheet magnesium. Its weight was reduced from 58 to 27 pounds . . . with every pound removed adding to the ease and efficiency of doing its work. It is a basic demonstration of magnesium in use . . . lightening the load for industry.

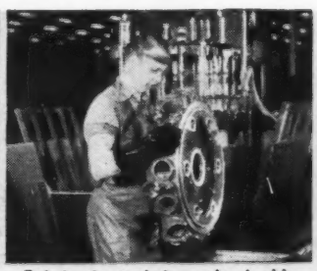
Today, in both consumer and industrial fields, magnesium is saving weight, increasing efficiency, without sacrifice of strength. For full information on its many advantages, contact the nearest Dow office.

Ready . . . to make products move!

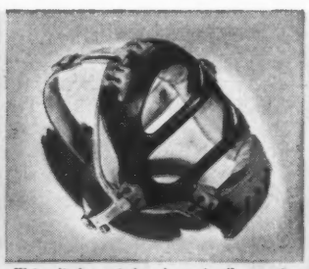
MAGNESIUM

LIGHTEST OF ALL STRUCTURAL METALS

Full stocks of magnesium in all common forms are now available from Dow, pioneer producer and today's foremost fabricator.



Fabrication techniques, backed by 30 years of Dow experience, are established in progressive plants throughout the country.



This lightweight baseball mask, in use in both major and minor leagues, was achieved through magnesium application.

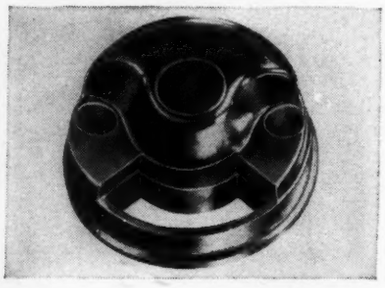
MAGNESIUM DIVISION • THE DOW CHEMICAL COMPANY, MIDLAND, MICHIGAN

New York • Boston • Philadelphia • Washington • Cleveland • Detroit • Chicago • St. Louis • Houston • San Francisco • Los Angeles • Seattle

of handling and lower costs. Having an adhesive back, the gasket does not need to be tied, laced or taped to vertical or inverted flanges; it can be finger-pressed onto the surface and will "stay put". Adhesive back gaskets have a crinolin liner upon which the template may be laid and either scribed or marked with an ordinary lead pencil. Quantities of gaskets may be cut at one time with the liner left in place. Chrome Lock gasketing is available in four types: Nos. 8 and 16 without adhesive back and Nos. 8A and 16A with adhesive back, both packed in either rolls or slit rolls.

Metering Valve and Slide

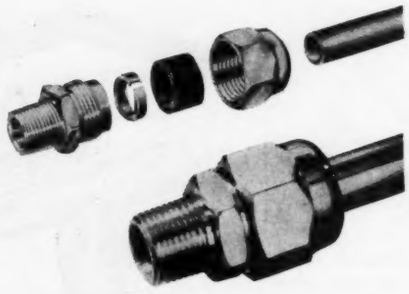
APPLICATIONS OF the new valves and slides produced by Morganite Brush Co., 3304 Forty-eighth Ave., Long Island City 1, N. Y., include all types of liquid, gas or viscous installations. The illustrated metering valve is



immune to acid, alkali and most solvents. Coefficients of expansion can be held to 0.000,001 and the valves will not warp or stick even in gum or hardening cold. Shear strength is to 10,500 psi; compressive, to 52,000 psi; and tensile, to 4000 psi. According to the company, good performances can be attained at 850 F.

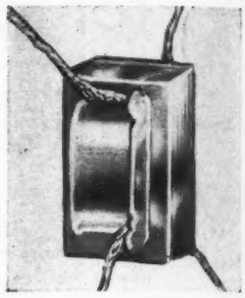
Flexible Tubing Fitting

MADE IN STANDARD sizes from ⅛ to 1½ inch outside diameter, the Flexigrip fitting offered by Gustin-Bacon Mfg. Co., Kansas City 7, Mo., consists of four parts: Body,



gripping ring, synthetic rubber gasket and nut. To attach the fitting, the nut, with gasket and ring inside, is slipped over any plain-end tube, cut to desired length. Tubing end is inserted into the body and the nut is tightened, which compresses the ring into a tight grip and molds the gasket around the ring for a leakproof seal. The seal is flexible and will withstand unusual vibration or impulse. Fittings are furnished in brass, aluminum or steel. Claims made by the manufacturer include the elimination of flaring, swedging, or soldering, by which means the tube adds to the strength of the joint, as well as saving time and labor in assembly.

Miniature Transformers



THUS FAR USED only in military equipment where light weight and volume were required, a series of "sub-ouncers" are being offered to industry by United Transformer Corp., 150 Varick St., New York 13, for use in hearing aids, vest pocket radios and other consumer equipment. These small transformers are 9/16 x ⅝ x ⅞-inch, and weigh 1/3-ounce. Coil is of layer-wound wire on a molded plastic bobbin. Insulation is cellulose acetate. While the bulk of this miniature unit production is to manufacturers' requirements, five standard types are available for immediate delivery.

Predetermined Electronic Counter

ANNOUNCED BY Potter Instrument Co., 136-56 Roosevelt Ave., Flushing, N. Y., a new electronic counter is for use in slide fastener manufacturing, counting and packaging of small items such as pills and buttons, and in rolling

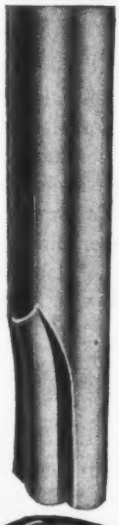
mills. The instrument employs four standard 4-tube counter-decade circuits arranged to give two independent predetermining channels in which any number from 0 to 10,000 may be set up by rotary switches on the front panel. Each channel is alternately preset to the desired predetermined number, accomplished automatically by self-contained circuits. This occurs in less than one millisecond. Input is arranged with either make-contacts or with sharp negative pulses, with frequencies in excess of 1000 cycles per second. Output consists of an ultra-high-speed relay with single-pole double-throw contacts. This relay is energized at the end of one predetermined count

WITH CLUTCH HEAD SCREWS

Norge

"Steps Up Production and Eliminates Cabinet Damage"

TYPE "A" ASSEMBLY BIT



COMMON SCREWDRIVER

Here Are Results computable in dollars and cents on the profit side of production . . . results obtainable only with CLUTCH HEAD Screws because they have exclusive features for safety and speed unmatched by any other screw on the market.

In the great Norge Plants, where the drive of screws runs into millions, safeguarding the beauty of an immaculate cabinet ranks in importance *as a cost factor* with speed of assembly.

With CLUTCH HEAD Screws, costly damage caused by driver slippage is eliminated:

1. Because the Center Pivot Column on the driver guides the bit into deep dead-center of the Clutch recess for automatically straight driving free from canting.
2. Because the driving engagement is all-square, eliminating the need for strenuous end pressure to combat "ride-out" as set up by tapered driving . . . one of the major causes of driver slippage.

The step-up in driving tempo logically results from the roomy easy-to-hit Clutch target, the positively centered entry, the sureness of the torque grip, and the effortless drive home. CLUTCH HEAD also contributes unmatched tool economy with the rugged Type "A" Bit that drives thousands more screws without interruption.

Note how simple it is to recondition this bit. It may be *repeatedly* restored to original efficiency by a 60-second application of the end surface to a grinding wheel. No "back-to-the-factory" shipment necessary. . . . No delay. No expense.

CLUTCH HEAD · SCREWS ·

Norge too, in common with all users of CLUTCH HEAD, enjoys the benefit of simplified field service because this screw is basically designed for operation with the ordinary screwdriver or any flat blade reasonably accurate in width.

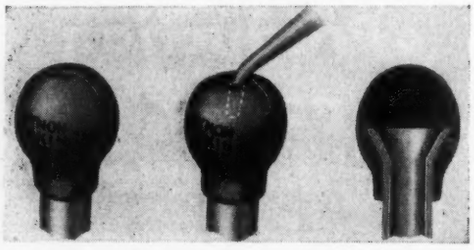
UNITED SCREW AND BOLT CORPORATION

CLEVELAND 2 CHICAGO 8 NEW YORK 7

cycle and remains energized throughout the other predetermined count cycle. Power for operations is obtained from a 110-volt 60-cycle circuit. The instrument is available for any specified total count with either the single or dual predetermining channels.

Self-Sealing Oil Tube Caps

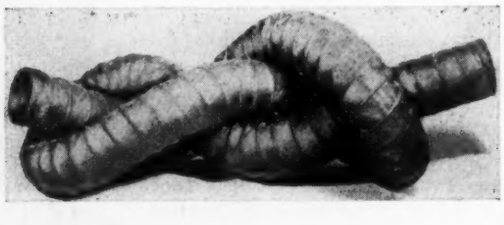
SELF-SEALING CAPS for oil tubes are tough, flexible and oil-resistant. Available in colors of molded synthetic Buna "N", the caps permit one-hand oiling from any direction. They are applied to the slightly flared end of an oil tube or oil line, and are permanently fastened by a combination of mechanical interference fit and synthetic ce-



ment. Caps are positioned by an internal shoulder, resting on the flared end of the tube. The cups hold a supply of oil for slow seepage to bearings. Tight sealing of the caps excludes dust and dirt. According to the manufacturer, Eynon-Dakin Co., 1847 West Bethune Ave., Detroit 6, there is no limitation as to size, the caps being presently available for oil tube sizes from ⅛ to ½ inch outside diameter.

Flexible Retractable Tubing

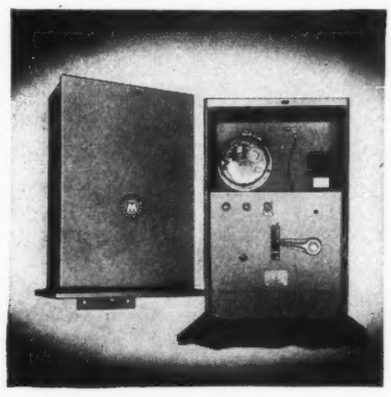
NONCOLLAPSIBLE under plus or minus pressures and retractable to about one-eighth its expanded length, a new type of flexible tubing known as Spiratube has been an-



nounced by the Warner Brothers Co., Spiratube Division, Bridgeport, Conn. The spring steel helix core causes it to spring out, like a jack-in-the-box, to its fully extended length, in which position it will stay regardless of working on pressure or suction. A feature of the construction is the method of spiral-stitching the spring core within the fabric. Inside surface is free of wire ridges providing little resistance to air flow and no obstruction to the passage of solids. Sharp bends may be made with only slight reduction of air flow and without use of elbows or special fittings. Spiratube is light and takes up little space when retracted. It is made of long-fibre duck having a bursting strength of 170 psi. The fabric is processed fire-resistant and covered with thermoplastic. For special applications a variety of other fabrics can be made to specifications. The tubing is furnished for portable or semipermanent ventilation or any handling of air, gases or light solids, in standard diameters from 3 to 16 inches and in lengths of 10, 15 and 25 feet. Built-in couplings permit quick joining or disconnecting.

Oil Field Pump Control

ALTERNATING-CURRENT magnetic oil-field pump controls, developed by Westinghouse Electric Corp., Pittsburgh, combine circuit protection and motor control in a single unit to save space and wiring. For automatic starting and stopping of oil field pumping motors, the starters are available in two basic types, each providing



overload and low-voltage protection. The automatic unit includes breaker, linestarter, lightning arrester, hand reset low-voltage relay, selector switch to permit manual operation, socket and time switch. The time switch automatically controls specific pumping periods during the day. The local control unit includes breaker, linestarter, lightning arrester and start-stop pushbutton. Ratings are from 1 to 25 horsepower, 110, 220, 440-600 volts.

Center-Tap Resistors

WIRE-WOUND resistors with center tap have been announced by Techtmann Industries, Inc., 828 N. Broadway, Milwaukee 2, distributors for Regan Engineering

Corp. The resistor consists of a core, machined from a solid block of steatite, upon which is wound a helical re-

Weld Penmetal Steelmesh

TO LIGHTWEIGHT METALS FOR HEAVY-DUTY WORK

Lightweight aluminum and stainless steel are used for heavy-duty work by welding STEELMESH to the sheets for reinforcement. An example is the successful application of flat rolled steel reinforced with STEELMESH for the fuselage, wings, tail surfaces and cowlings of airplanes. In a given area of STEELMESH, only 10% is solid steel (the remainder is interstices). It is, therefore, possible to spot weld this mesh to a thin sheet and secure a composite construction much stronger than a solid steel sheet of the same weight.

The strength imparted by STEELMESH is due to the trusslike formation of the mesh and to the fact that in the manufacture of this product, the openings are simultaneously slit and cold drawn or expanded.

STEELMESH is used in the construction of bulkheads, open partitions, insulation support, guards and ventilators, lockers, cages, truck bodies, shelving, racks, baskets, drying trays, gratings, air filters, drainers, storage bins, grille work, screening, mastic flooring, dehydrators, toys, novelty work and for many other purposes.

Write for catalog and samples.

PENN METAL COMPANY, INC.

Dept. 30, Parkersburg, W. Va.

District Sales Offices

BOSTON • NEW YORK • PHILADELPHIA • CHICAGO • DETROIT • PARKERSBURG, W. VA
LOS ANGELES • SAN FRANCISCO • DALLAS

METAL MARK Penmetal 1869 OF QUALITY

sistance wire element. Center and end taps are half-straps securely clamped to the core, assuring mechanical rigidity independent of support by the resistance element. The finest wire can be tapped off with positive assurance of a good, nonheating connection. The resistor is so designed that should a unit be overloaded and burn out, the coil can be removed and replaced in a few minutes. Connections or supporting assembly need not be disturbed.

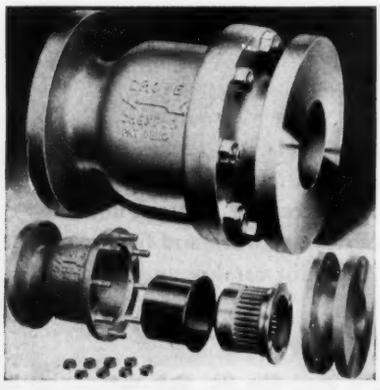
Wire Rope Clamp

DEVELOPED BY the Nunn Mfg. Co., 2125 Dewey Ave., Evanston, Ill., a new Cabl-Ox clamp incorporates a wedging action in its component parts, making it possible

to hold loads in excess of the tensile strength of the rope used. The unit is alloy steel, cadmium plated for weather protection. Being easily assembled and disassembled results in saving of time where wire rope is used. Standard sizes in which the clamp is available range from 1/16 to 3/4-inch.

Flow Check Valve

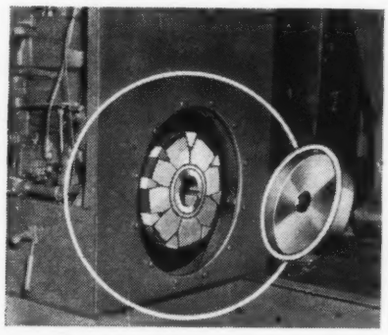
BECAUSE OF the self-compensating and wear-resistant factors, a new check valve announced by Grove Regulator Co., 6505 Green St. at 65th, Oakland 8, Calif., is particularly suited for handling highly corrosive and erosive air, gases or liquids. Operation is effected by a synthetic rubber tube, stretched over a slotted, cup-shaped metal



core, which expands to open and contracts to close. Inasmuch as the expansible tube closes on balanced flow, prior to commencement of back flow, there is no tendency to establish pressure impulses, shock or water hammer through the flow line. The flexive tube possesses a uniform and constant spring rate which avoids inertia being set up to cause operation beyond flow requirements.

Centrifugal Clutch

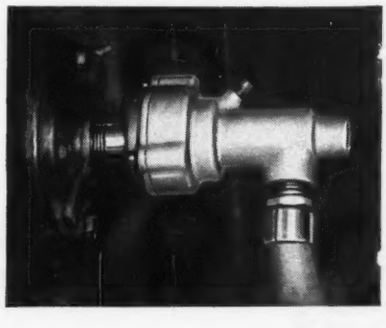
MANUFACTURING and sales rights have been obtained by the Hardinge Co. Inc., York, Pa., from the Automatic Corp. of Canada, for its Auto-Centri clutch. Some of the advantages to be gained from the installation of these clutches to electric motors are: A squirrel-cage motor can be used, alternating-current motors can be started across the line; clutch permits motor to come up to the speed at which it develops its maximum torque so that it is available for starting purposes, and damage from overloading is eliminated. Consisting of three main parts—drive body, driven body and driving mechanism—the clutch is available in sizes from fractional horsepower to 5000 horsepower for any type of power drive. The interlocking feature of the driving blocks insures that the metal drive blocks move in unison, resulting in balanced delivery



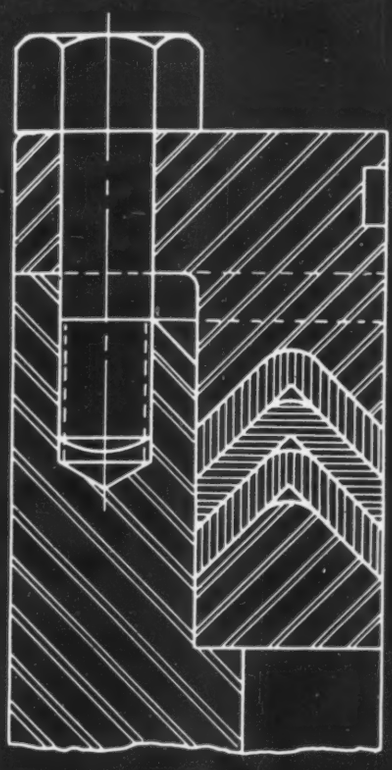
of power. The connecting elements between each pair of drive blocks rest against the bosses on the face of the drive body, preventing blocks from contacting the surface of the friction wall. Thus even if linings are worn, no damage can occur to the clutch.

Rotating Water Joint

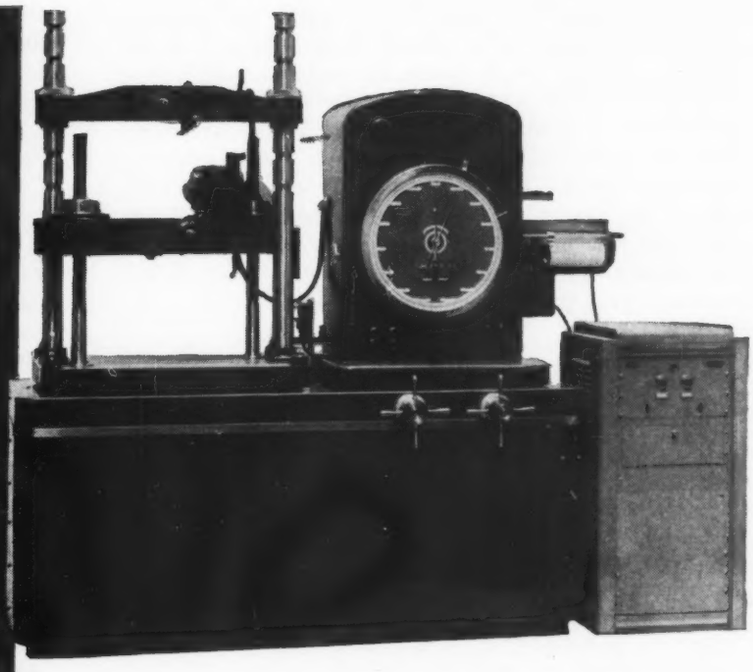
FOR COOLING revolving shafts, cylinders or drums, a new type rotating water joint now being marketed by Deublin Co., Northbrook, Ill., will also handle other liquids up to 200 F. Leakproof, corrosion-proof and frictionless



the joint utilizes a helical groove on the rotor which runs counter to the direction of rotation. The sweeping action



Section of gland and bottom piston used in Olsen Hydraulic Testing Machine, showing VIM Leather "V" packing installation.



Controlling the power that tears steel apart

Vim Leather "V" Packings are an essential item in the Olsen Hydraulic Universal Testing Machine.

These standard testers, equipped with electronic recorder shown at right on above photo, exert forces ranging from zero to 160,000 pounds. It tears steel to shreds, and records the stress-strain curve, magnified 500 or 1000 times on the chart.

Olsen has long depended on VIM Leather. The unit above uses resin-impregnated "V" packings, the same type developed for aircraft hydraulic mechanisms. These packings were pretested by Olsen to prove their ability to hold the enormous pressures exerted by this unit.

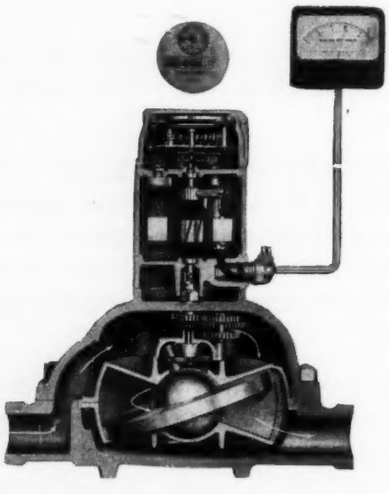
Another evidence of the universal acceptance of VIM Leather Packings! Write for our abbreviated catalog, or for any data needed from our packing engineers. E. F. HOUGHTON & CO., 303 W. Lehigh Ave., Philadelphia 33, Pa.

HOUGHTON'S *Engineered* **VIM** *Leather Packings*

of the helix prevents leakage between shaft and housing, as well as keeping foreign objects from lodging between the two surfaces. A maximum clearance of 0.0005-inch has been provided between the shaft and housing. Rotor concentricity is held to 0.001-inch total indicator reading. Both the machined stainless steel rotor and the bronze housing resist the corrosive action of water, brine and other similar liquids that may be used for cooling or heating. The new union is offered in two types: Monoflow and Duoflow, in four standard sizes. Other sizes can be produced from ¼-inch pipe size to 6 inches.

Indicating Flow Meter

FOR applications such as measuring, indicating, and totalizing the flow of oil, water, lacquer, tar, molasses and other unusual liquids, the Hays Corp., Michigan City, Ind., has announced its Veriflow meter. The outstanding feature of the meter is its ability to indicate the rate of flow at a point remote from where the meter is installed. This is in addition to the rate of flow indication and integrator which are integral with the meter itself. To secure the remote indication the generator is mounted on top of the flow meter, the same shaft operating both the generator and the totalizing register. No external source of electricity is required because the indicating



meter measures output of generator and is calibrated to indicate the corresponding rate of liquid flow through the meter. The scale can be calibrated to read in any values of flow desired such as gallons per minute.

Hermetically Sealed Resistor

TO THE LINE OF hermetically-sealed Akra-Ohm fixed wire-wound resistors made by the Shallcross Mfg. Co., Jackson and Pusey Aves., Collingdale, Pa., a new small-size unit rated at 0.5-watt has been added. Known as Type 1101, the new resistor is ⅞-inch long and ⅞-inch in diameter, and is designed for style RB12A under JAN Specification R93. Maximum resistance value when wound with nickel-chromium wire is 350,000 ohms, maximum voltage 420. Resistance element, winding form and protective ceramic shell form a rigid, integral unit. Hermetic-sealing is obtained by a special solder process.

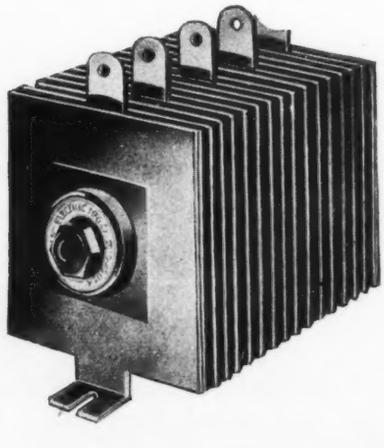
Stainless Seamless Tubing

SEAMLESS FLEXIBLE tubing in stainless steel is now available from The Seamlex Co. Inc., 27-27 Jackson Ave., Long Island City, New York, in standard sizes from 1/16-inch inside diameter to ¼-inch inside diameter, inclusive. One end of the new stainless tubing shows the deep helical

convolutions in cross section; the other illustrates an integral cylindrical extension suitable for attaching flare-type fittings. The tubing shown is usually provided with metallic braid jacket to take care of internal pressure requirements. Any type of end connection can be furnished.

Self-Cooling Rectifier

DEVELOPMENT OF a dry disk metallic rectifier which does not require forced cooling has been announced by The Benwood-Linze Co., St. Louis. The rectifier is rated at 50 amperes for 6-volt automotive battery taper charging. Two units may be operated in parallel from separate transformer secondaries, providing 100 amperes maximum charging rate without a fan. Chargers of this type op-



erate from the usual 110 volt a-c power supply, to deliver the required rating of direct current to the battery.

How to Reduce 25 Lubrication Points to ONE!

Illustration shows an installation of an Alemite LubroMeter Centralized Lubrication System on a large stamping press. This system handles either oil or grease—can be designed into almost any type of machine.

By itself, an Alemite System appears to be composed of "metal arteries" and valves. Actually, it's the "lifeline" of a machine—your machine, if you like.

Maybe your machine has 10 lubrication points—or 25, or 40, or 60. With an Alemite System there'd only be *one*—the one central point where the lubricant is introduced. From that *one* point *all* points would be positively lubricated with a metered quantity of grease or oil without *stopping the machine!*

Results like these are common. A huge rotogravure printing press had to be shut down 2 hours daily for lubrication, and bearing failures were frequent. Today, with an Alemite Centralized System, the press is positively lubricated, while in operation, in only 10 minutes per 24 hour day. *And*—bearing failures have ceased to exist.

When you design Alemite Centralized Lubrication into your machines you automatically protect their performance from the dangers of human error. You reduce the number of lubrication points to *one*. You guarantee positive lubrication without "time-outs." You cut repairs to a minimum...lengthen machine life... increase satisfaction. It all adds up to More Productive Time per machine. And that enhances your reputation as a designer.

* * *

Without obligation, have an Alemite Lubrication Specialist demonstrate one or all 4 new Centralized Systems right at your desk with transparent working models. Also, ask for any technical help you want regarding Alemite Systems. Drop a note on your letterhead to Alemite, 1804 Diversey Parkway, Chicago 14, Illinois.

ALEMITE

REG. U.S. PAT. OFF.

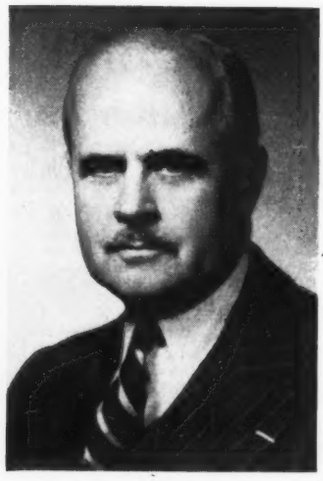
Only Alemite Combines ALL 3 in Lubrication

1. EQUIPMENT 2. PROCEDURES 3. LUBRICANTS

Another Product of STEWART WARNER

L. A. Philipp

Ralph M. Heintz



Harry A. Toulmin

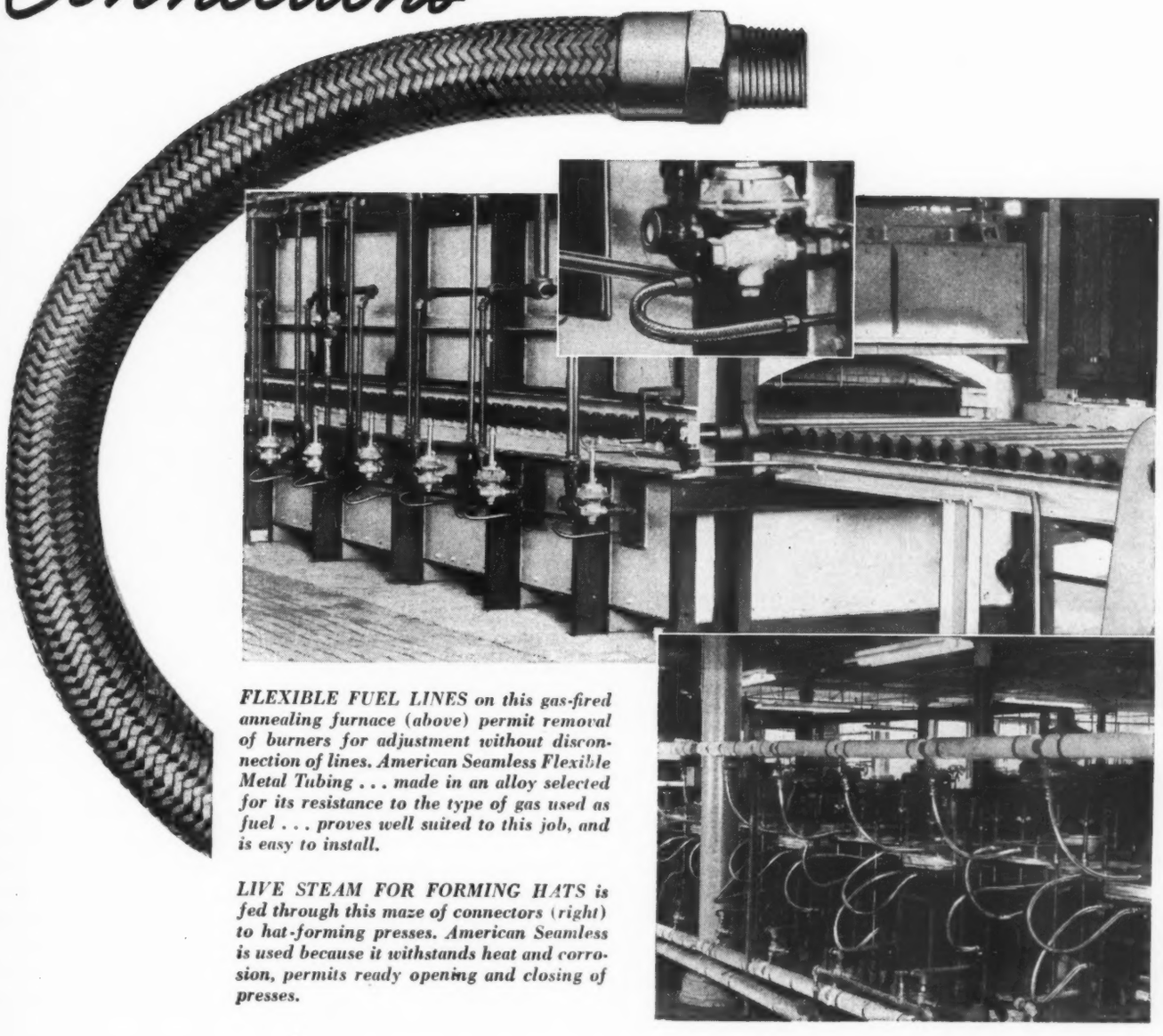
MEN... of machines

DR. L. A. PHILIPP, recognized as one of the outstanding men in technical research, has been elected vice president in charge of engineering, Kelvinator Division, Nash-Kelvinator Corp. Dr. Philipp joined Kelvinator in 1927 as director of research, after lengthy research fellowships at the University of California and the University of Michigan, where he received his doctor's degree in 1925. In 1937 he was appointed manager of the Detroit plant, and two years later became chief engineer at Kelvinator, the position he has held prior to his present election. Dr. Philipp has participated in many important refrigeration and appliance developments and is believed to hold more refrigeration patents than any other individual. His recent citation for his contribution to the war effort, given by the Air Technical Service Command reads: "Dr. Philipp's keen analysis and diligent efforts in developing an efficient testing instrument in connection with Nash-Kelvinator's program to manufacture Hamilton Standard type governors brought about a solution to a major production problem at a time when there was no comparable testing equipment available for speedy production checking."

RALPH M. HEINTZ, vice president and secretary of Jack & Heintz, has been named vice president in charge of engineering of the newly formed Jack & Heintz Precision Industries Inc. Because his father, Dr. Heintz, was connected with the mining industry, it was natural for Mr. Heintz to become interested in mining and to be a mine foreman at the age of 18 and a superintendent at 19. Returning to school, he was graduated from California School of Mechanical Arts. Later he attended the universities of California and Stanford. During World War I he spent eight months overseas as a private in the Signal Corps (Air Section), doing flying navigation work and radio research. After the war he again took up his studies, graduating in 1920. He then joined Standard Oil Co. and, after eight months, started his own organization, Heintz & Kaufman Ltd., building scientific apparatus and wireless equipment. It also formed the Globe Wireless Ltd., comprising of nine transoceanic and coastal stations and a fleet of thirty ships. Between 1935 and 1937 Mr. Heintz worked on the development of high-frequency a-c power plants for Army aircraft and radio intercommunication and accessory equipment. In 1940 he became a partner of Jack & Heintz Inc. which designed and built the JH 5 and JH 10 engines.

HARRY A. TOULMIN, recently re-elected president, also is general manager and chairman of the board of The Hydraulic Press Mfg. Co. Born in Springfield, O., in 1890, Mr. Toulmin was admitted to the Ohio bar in 1913,

Flexible Connections to FURNACE and PRESS



FLEXIBLE FUEL LINES on this gas-fired annealing furnace (above) permit removal of burners for adjustment without disconnection of lines. American Seamless Flexible Metal Tubing . . . made in an alloy selected for its resistance to the type of gas used as fuel . . . proves well suited to this job, and is easy to install.

LIVE STEAM FOR FORMING HATS is fed through this maze of connectors (right) to hat-forming presses. American Seamless is used because it withstands heat and corrosion, permits ready opening and closing of presses.

HEAT, vibration, pressure, movement, abrasion or corrosion may very well raise havoc with the type of flexible connections you now employ. If so, look into the possibility of doing a better, safer, more economical job . . . with American Flexible Metal Hose or Seamless Flexible Metal Tubing.

Specially engineered assemblies of these durable products have solved serious problems for countless other manufacturers. The applications illustrated are typical. For detailed information, write for Publication SS-50.

48545

ANACONDA

American

METAL HOSE

THE AMERICAN BRASS COMPANY

American Metal Hose Branch

General Offices: Waterbury 88, Connecticut

Subsidiary of Anaconda Copper Mining Company

In Canada: ANACONDA AMERICAN BRASS LTD., *New Toronto, Ont.*

later to the U. S. Courts and in 1919 to the Supreme Court. Prior to World War I he was connected with Recording and Computing Machines Co., solving mass production and engineering problems. In 1917 he was appointed assistant secretary of the General Munitions Board of the Council of National Defense and later was made executive, gun division, Ordnance Department, U. S. Army where he supervised all operations from design to final production.

As chief of aircraft armament of the American Expeditionary Force early in 1918 he was responsible for machine gun manufacture in France and the design of armament equipment for aircraft and its installation. He completed his war service after serving as chief of staff of the Army Air Service. In the interval between World War I and II he has been a director, executive, or concerned with engineering and production of different types of heavy machinery, as well as an inventor of both heavy and light machinery. During World War II his responsibility not only concerned design, but also production and distribution. He was charged with the design and manufacture of floating cranes and port freight handling equipment as well as the design and production of all locomotives and cars used by the United States and Allied Nations. Prior to returning to Hydraulic Press he had been connected with the Air Service Command.

♦

WILLIAM W. HENNING, formerly engineer in charge of design analysis, International Harvester Co., Chicago, has been appointed engineer in charge of crawler tractors for the company.

♦

CHARLES W. BUCKLEY, senior design engineer previously with the Ford Instrument Co., and LT. COMDR. F. RAY LINDA, USNR, former design and development engineer with the Bemis Bag Co., have been added as research engineers to the staff of Wright's Automatic Machinery Co.

♦

WILLIAM WISEMAN who had been chief engineer of Warner Aircraft Corp., Detroit, has joined Continental Motors Corp. as assistant chief engineer of the aircraft division.

♦

MAJOR F. R. NAIL is assistant to the chief engineer in charge of highway truck developments, Mack-International Motor Truck Corp. Major Nail had been assistant chief, Engineering Branch, U. S. Army, Office of the Chief of Ordnance, Detroit.

♦

CLARENCE H. SAMPLE has been appointed chief engineer of Rheem Research Products Inc.

♦

CARL J. SNIDER has joined Greer Hydraulics Inc. as chief engineer in charge of development of aircraft hydraulic testing machines and maintenance equipment.

♦

JAMES A. SCOTT, formerly lubricating engineer, Bendix Aviation Corp., New York City, has been appointed director of research and development, Marshall-Eclipse Division, Bendix Aviation Corp., Troy, N. Y.

♦

ALEX PETROVSKY has become connected with Republic Aviation Corp., Farmingdale, L. I., N. Y., as aerodynamics engineer. He had been with the National Advisory Committee for Aeronautics, Langley Field, in the same capacity.

♦

RALPH E. MIDDLETON, vice president, Aireon Mfg. Corp., has been named general manager of the corporation's hydraulic division at Burbank, Calif. Mr. Middleton had been chief engineer there since 1941.

Aluminum Springs

(*Concluded from Page* 140)

springs, calculated in terms of *per cent loss of load*, is shown by the following table for springs that were continuously subjected to compressive stresses for one year:

Loss of Load After Continuous Stresses for One Year

Applied Stress (psi	Loss of Load (%)
16,000	0.0
24,000	0.0
35,000	1.7
43,000	2.8

Probably the greatest single factor accounting for complete failure of springs in general is fatigue and, therefore, this characteristic was carefully studied in R303. The test commonly used to determine the fatigue or endurance limit of a material is the rotating beam fatigue test. In this test, as determined by the R. R. Moore and the Krouse machines, the endurance limit of R303 was 23,000 psi at 500,000,000 cycles and 24,000 psi at 100,000,000 cycles of stress reversals. The same numerical value for the endurance limit was found for both the T275 and T315 tempers.

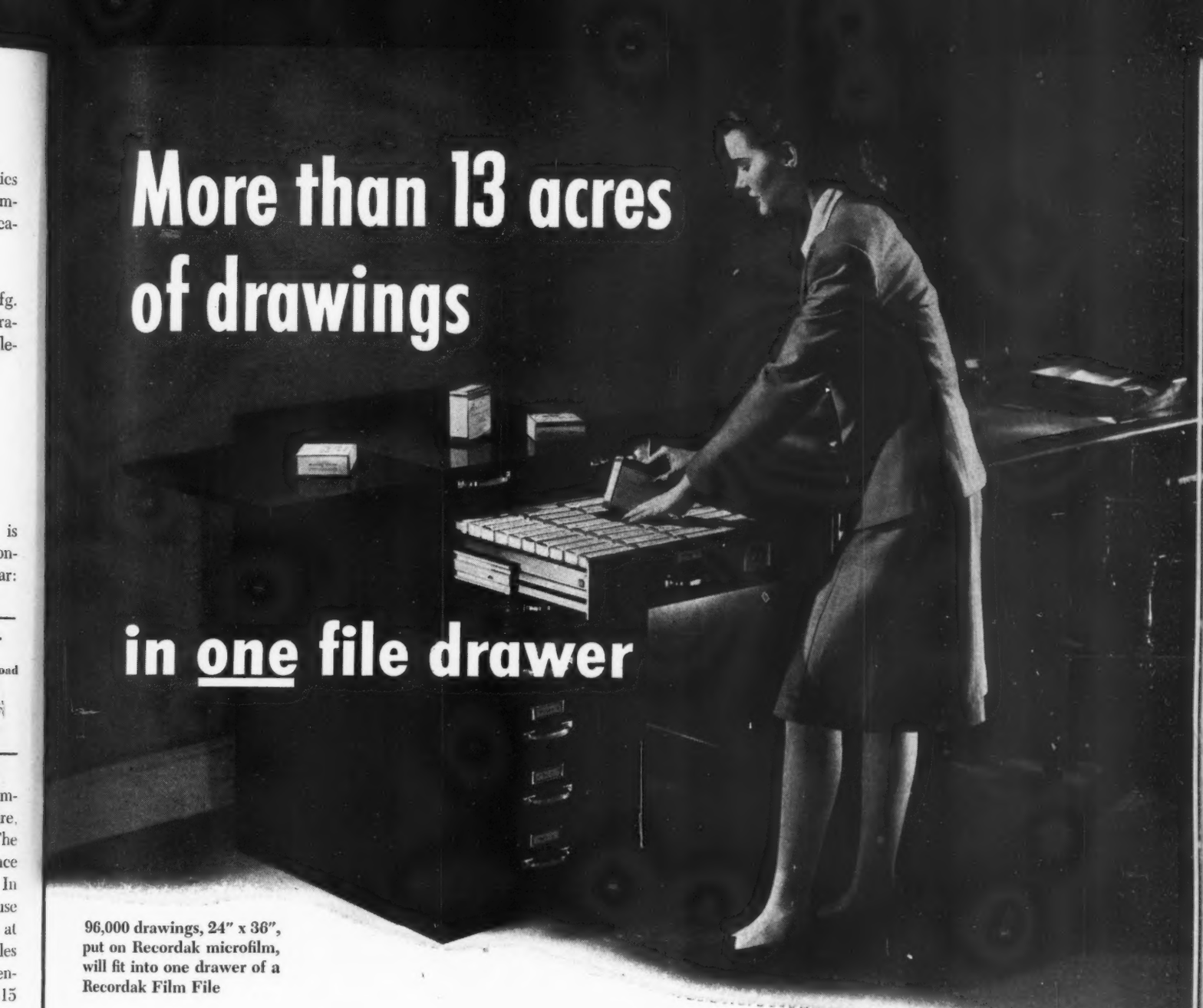
However, since the standardized fatigue test utilizes a polished specimen, and is, therefore, not a true representation of the surface conditions with its various stress raisers that may be present in the fabricated product, the springs themselves were subjected to repeated stresses in specially constructed fatigue machines. Results of fatigue tests on R303 springs are shown in *Fig.* 4. As in the standardized fatigue test, no appreciable difference in resistance to fatigue failures could be detected between springs wound from T275 wire and those wound from T315 wire.

The endurance of aluminum springs is materially improved by controlled surface peening. Shown also in *Fig.* 4 is the fatigue life of similar springs that have been peened to an intensity of 0.011A2 (Standards of General Motors Research Laboratories) using shot 0.0275-inch in diameter under 25 pounds pressure at a ⅜-inch nozzle orifice. For peened and unpeened aluminum springs, the respective endurance limits are 17,000 and 10,000 psi at 10 million cycles; 27,000 and 12,000 psi at one million cycles.

Grateful appreciation is extended to Mr. R. W. Helmig of the W. W. Sly Manufacturing Co., Cleveland, for surface peening some of the springs; and to Messrs. L. M. Puster, D. L. Morgan and D. F. Wolfe of the Fulton Sylphon Co. (Subsidiary of the Reynolds Metals Co.), conducting fatigue tests on the springs.

More than 13 acres of drawings

in one file drawer



96,000 drawings, 24" x 36", put on Recordak microfilm, will fit into one drawer of a Recordak Film File

MILLIONS of bulky war drawings must be kept several years . . . a serious filing problem for industry . . . yet one that can be solved quickly and easily with Micro-File Recordak.

Government regulations permit the destruction of microfilmed World War II contract originals, subject to certain conditions (Contract Settlement Regulation No. 11, January 31, 1945). This means you can actually "debulk" your war files by 98% . . . use the space gained for better purposes. You can get hundreds of big engineering drawings . . . thousands of small ones . . . on one 100-foot roll of 35mm. Recordak microfilm.

You protect your micro-size files against misfiling and tampering. You can store them in fire- and theft-proof vaults. Yet they are right at your fingertips for reading or copying. You can make full-size facsimile prints quickly—in any quantity you want—with a Recordak Enlarger.

New book answers your questions

Write for the new, free book on microfilming—"50 Billion Records Can't Be Wrong." It tells you about the many applications of Recordak. Also ask about the nation-wide customer service Recordak maintains. Recordak Corporation, Subsidiary of Eastman Kodak Company, 350 Madison Avenue, New York 17, New York.

Mail this coupon today!

RECORDAK

(Subsidiary of Eastman Kodak Company)

originator of modern microfilming —and its uses in industry

Recordak Corporation,
350 Madison Ave., New York 17, N. Y.

Please send the new book about Recordak, "50 Billion Records Can't Be Wrong."

Name_____

Company_____

Street_____

City and State_____

19

Construction of Nomographs

(*Continued from Page* 126)

required, since the equation of the scale is no longer $y = R$, but $y = log\ R$. Therefore, a new line is drawn to the right of R and is called the S scale. These scales should not be drawn too close together, or accuracy is impaired. If this scale should not exceed 4 inches,

$$m_1 = \frac{4}{log\ 100 - log\ 1} = \frac{4}{2} = 2$$

The h scale should range from 1 to 12 and have a length of 5 inches

$$m_2 = \frac{5}{log\ 12 - log\ 1} = \frac{5}{1.0792} = 4.64$$

therefore

$$m_3 = \frac{9.28}{6.64} = 1.4$$

Since $a - b$ was chosen to be 3 inches

$$\frac{b}{3} - \frac{4.64}{6.64} b = 2.1 \text{ inches}$$

The S scale can be graduated from 0 to 100 and equal points on R and S scale connected (for example, points 1, 2, 5, 10, 20, 30, 40, 50, 60, 70, 80, 90 and 100). These lines will serve as guide lines on the interchange from natural to logarithmic scales. Then the h scale can be completed, and one point on the W scale can be located using the original formula. Also, the W scale, in turn, can be finished, its equation being $y = m_3\ log\ (x/c) = 1.4\ log\ (x/0.22)$. Since the shift of the W scale has been established by locating one point mathematically and graphically, the W scale can be designed with the formula $y = 1.4\ log\ x$.

To read the chart in the previous example, where $D = 10$ inches, $d = 5$ inches, $h = 5$ inches, lay a straight edge through points 10 and 5 and mark intersection on R scale. Then follow approximately parallel the nearest guide line to the S scale. From there lay the straight edge through point 5 inches on the h scale. The intersection on the W scale will give the weight, in this case approximately 82 pounds.

Thus, it can be seen that there isn't much of a trick to constructing alignment charts. Interpreting them is even simpler. Wider use of these time-saving mathematical tools in working out design problems would certainly be justified.

Improving Railway-Bar Bearings

CONSIDERING the geometrical perfection of the oil film designed into bearings for modern high-speed machinery, the behavior of the crudely constructed railway journal bearing is most impressive. Occasionally there might be an epidemic of hot-boxes but year in and out the general performance has been satisfactory. Hence, aside from economic considerations, there has appeared to be no urgent necessity for changes.

Regardless of its record, however, the railway-car journal bearing is susceptible of improvement. It is obvious that the frictional drag of the waste pack consumes power which could be used in hauling useful load. It is also clear that a better fit between journal and bearing would result in less load per unit of bearing area, increased oil-film thickness, and greater load capacity.

Minimum film thickness is the criterion of bearing safety. In operation a properly fitted bath-lubricated bearing is completely separated from its journal by an oil film of appreciable thickness, except at the moments of starting and stopping. Thus wear is reduced and bearing safety increased. These considerations are not new. Fitted journal bearings are common on European railway cars, and replacement of waste pack by bath lubrication has been made successfully abroad. The problem has also received some attention here.

To evaluate the possibilities in this direction, comparative tests of fitted and broached bearings lubricated by bath, pad, and waste pack were undertaken. Tests were made at 40, 80, and 100 mph. On the basis of present information, the following conclusions appear warranted:

1. Pad and waste pack are capable of supplying sufficient oil to maintain complete fluid-film lubrication at speeds at least as high as 100 mph.

2. The railway-car bearing is capable of carrying its maximum load without failure at temperatures at least as high as 375 F.

Fitted and Broached Bearings Compared

3. Operating temperatures and frictional power losses are somewhat less with the broached than with the fitted bearing. These advantages are offset by the fact that under similar operating conditions the oil-film thickness of the broached bearing averages only 1/3 that of the fitted bearing. Hence the broached bearing has less reserve load capacity and safety.

4. In a waste-pack-lubricated box, friction of the waste pack is approximately the same as the friction of the bearing.

5. Friction of the pad used in these tests is about 1/3 that of the waste pack.

6. Total friction with bath and pad lubrication are nearly the same, due to the higher operating temperature caused by the pad.

7. Total friction with waste-pack lubrication is higher than with bath or pad despite the higher operating temperatures due to the friction and blanketing effect of the waste pack.

8. Because of its lower operating temperature the bath-lubricated bearing has greater oil-film thickness, hence greater reserve load capacity and safety. This is particularly important at high speeds.

9. Frictional power loss and drawbar resistance of the waste-pack-lubricated bearing are approximately 1/3 greater than with bath or pad lubrication.

10. Method of lubrication has a profound effect on the heat-dissipation characteristics of the box and the temperature gradient between journal and box.—*From a paper by S. J. Needs, Kingsbury Machine Works Inc., presented at the recent ASME annual meeting in New York.*

For Greater Strength with Reduced Weight

J&L OTISCOLOY

SHEETS · PLATES

J&L STEEL

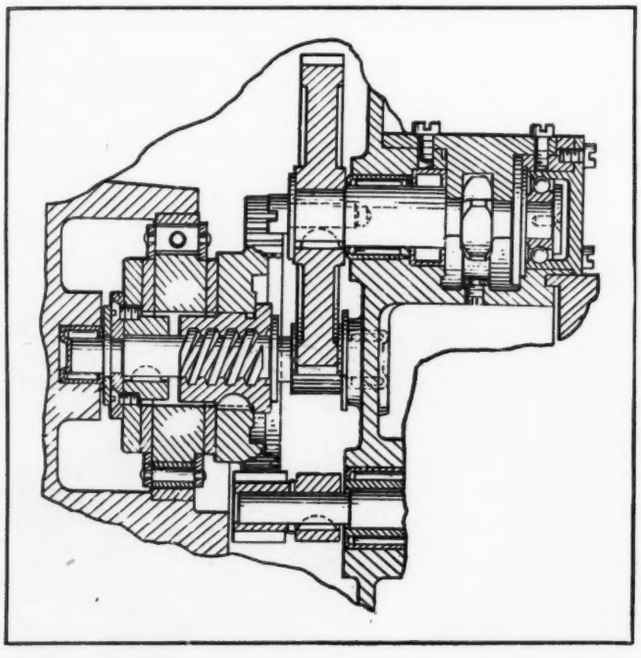
The inherent strength of Otiscoloy is obtained without mechanical working or heat treating, which gives it exceptional weldability and formability for a high-tensile steel. It is being used successfully to reduce dead-weight and increase pay load in railroad freight cars, in trucks, buses, mine cars and other equipment. Otiscoloy is also resistant to abrasion and corrosion. Our metallurgical engineers will be glad to discuss the application of Otiscoloy to your production problems. Write for further information.

JONES & LAUGHLIN STEEL CORPORATION

PITTSBURGH 30, PA.

Noteworthy Patents

UNIFORM DISTRIBUTION OF BRAKING LOAD is created automatically by the hoist load brake covered in patent 2,372,026 assigned to Manning, Maxwell & Moore, Inc., by Ralph E. Smith. The design utilizes a novel float-

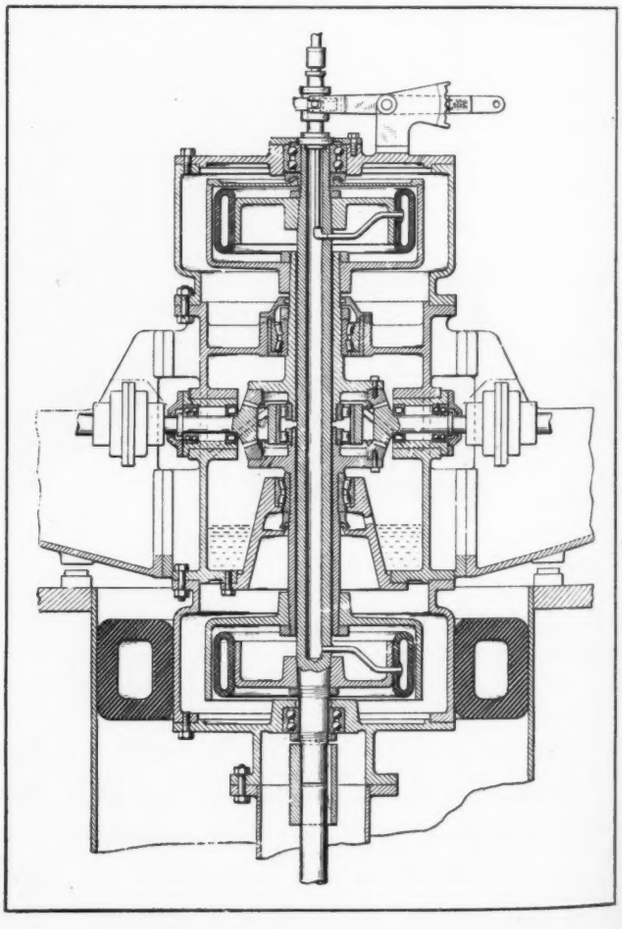


ing-roller ratchet mechanism to assure satisfactory roller life and positive braking action. The roller ratchet brake is used in conjunction with a screw-driven disk brake for checking or locking descent of a load.

ELIMINATION OF DESTRUCTIVE CHATTER is achieved by a pressure-regulating bypass valve for constant-delivery hydraulic pump systems covered in patent 2,362,713 assigned to International Harvester Co. by Carl W. Mott. An auxiliary control in this valve prevents creation of a bypass condition while oil is being supplied to a hydraulic motor in the system, irrespective of the magnitude of the pressure attained in operation.

COOLING OF LUBRICATING OIL for antifriction journal bearings is accomplished effectively by means of a hollow agitator member which is designed to dip into the oil reservoir as outlined in patent 2,372,896. Assigned to the Air Preheater Corp., by Sven Holm and John Vickland, the star-shaped hollow paddles of this agitator rotate with the shaft while a cooling or refrigerating fluid circulates continuously within them.

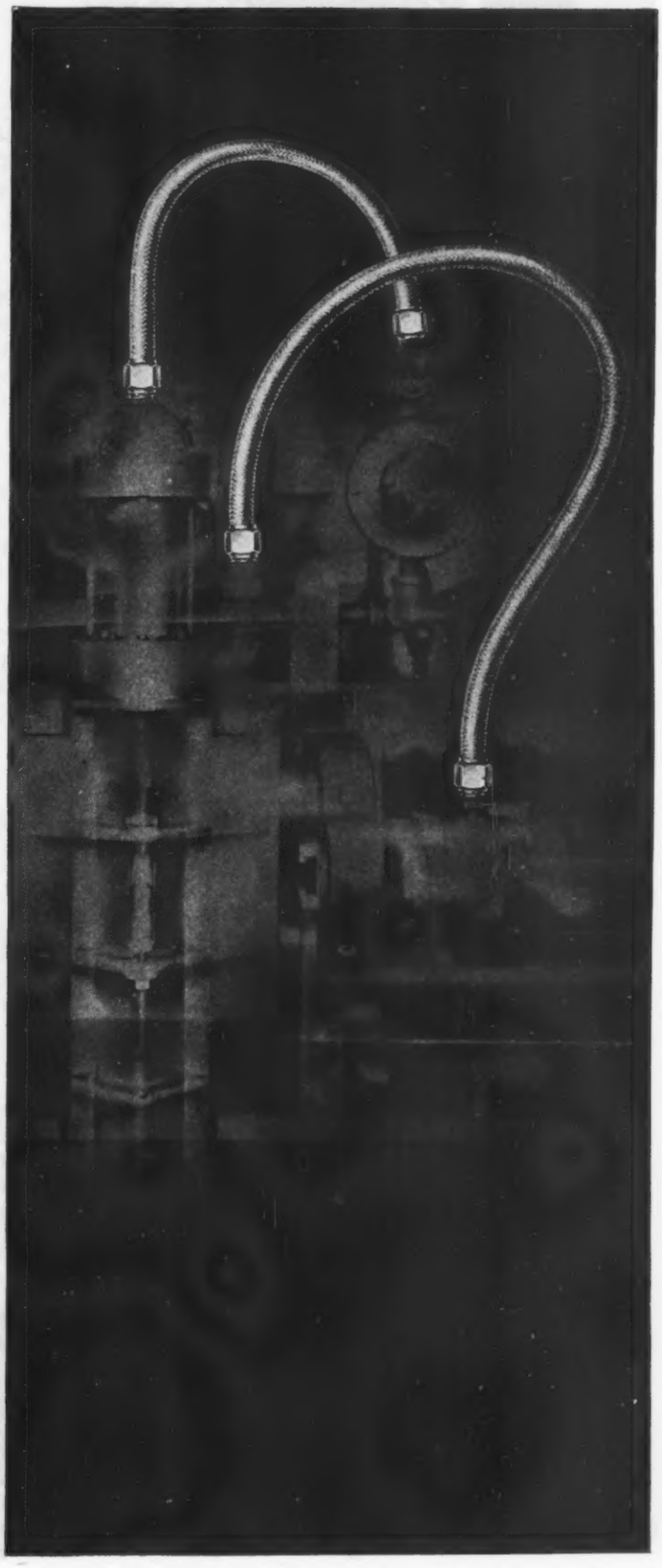
COMBINING A MULTIMOTOR SUPPORT and a self-contained power transmission, a compact unit covered by patent 2,368,885 is primarily intended for marine use. Assigned to the Falk Corp. by W. P. Schmitter, the design



provides for the utilization of power derived from a number of prime movers to drive a single propeller shaft in either direction. Tire-like annular glands are used to transmit the power without shock or vibration and similar pneumatic glands support the entire unit in place by frictional engagement permitting easy withdrawal and replacement.

PRESSURE-TIGHT PERFORMANCE FOR YEARS...

at 475 Oscillations per Minute!



You're looking at Titeflex all-metal, flexible tubing serving a double purpose. First, as a hydraulic oil line supplying up to 75 pounds p.s.i. Second, as the exhaust return line.

Primary reason for the use of Titeflex in this out-of-the-ordinary installation is that its inherent pressure tightness—despite constant oscillation—prevents the line leakage that would mean impaired operating efficiency...work spoilage.

Titeflex likewise provides identical trouble-free protection and service while operating under constant vibration, pressure, heat, cold, or corrosive attack.

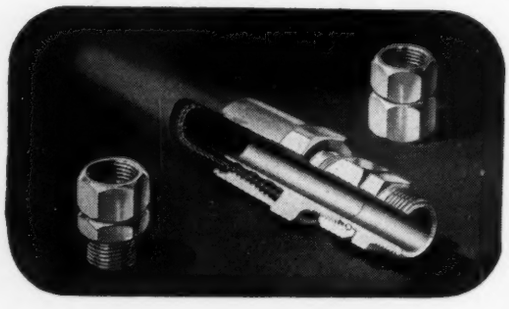
If you are not already aware of the economies and increased service performance that stem from Titeflex installations, our application engineers will gladly give you the facts. Titeflex, Inc., 532 Frelinghuysen Avenue, Newark 5, New Jersey.

Titeflex

THE ALL-METAL FLEXIBLE TUBING
THAT STAYS TIGHT

HOW TO AVOID *Costly* BREAKDOWNS!

TORSIONAL strains in flexible hydraulic control hose assemblies result in costly breakdowns. These strains which are ordinarily set up when installing hydraulic control hose assemblies are eliminated by the use of Eastman Adapter Unions. Maximum hose life is assured when Eastman Adapter Unions are used



ANOTHER tip: Eastman "Pressed-On" couplings, combined with adapter unions and high pressure wire braided hose, provide a leakproof and durable hydraulic control hose assembly. A complete Eastman assembly can always be depended upon, because the patented gripping feature of the Eastman "Pressed-On" coupling makes the bond of coupling to hose stronger than the hose itself. Be Sure! Specify Eastman Hydraulic Control Hose Assemblies with Adapter Unions.

For complete specifications, write

THE EASTMAN MFG. CO.

1002 N. 11TH STREET, MANITOWOC, WIS.

BUSINESS AND SALES BRIEFS

ASSOCIATED with the company for thirty-eight years, Walter F. Spoerl has been named general sales manager of the mechanical goods division of United States Rubber Co. He succeeds Herman A. Everlien who died recently.

♦

Lukenweld Inc., division of Lukens Steel Co., has added E. R. McClung Jr. to its staff as welding engineer. Mr. McClung will be responsible for procedures, techniques, training and new developments in welding. Formerly he was connected in a similar capacity with the New York Shipbuilding Corp. of Camden, N. J.

♦

Recently announced by Superior Carbon Products Inc., Cleveland, is the appointment of Theodore E. Liebert as sales manager. Formerly Mr. Liebert served as superintendent of production.

♦

To fill the vacancy caused by the sudden death of Horace B. Keeler last fall, Hydraulic Equipment Co. of Cleveland has elected Herbert P. Ladds to its board of directors. Mr. Ladds is president of National Screw & Mfg. Co. and of the Cleveland Chamber of Commerce.

♦

With headquarters at Amesbury, Mass., Alfred F. Gauron has been appointed field engineer of the New England district of Advance Pressure Castings Inc., Brooklyn, N. Y.

♦

Opening of a sales and service office at 2509 Commerce St., Dallas, Tex., has been announced by Fawick Airflex Co. of Cleveland. John V. Eakin, formerly clutch service manager at the Cleveland plant, has been named manager of the new office.

♦

Following the resignation of Max Giffey, Link Belt Co. has appointed Arthur J. Olson as district sales manager at Kansas City, Mo. Associated with the company since 1918, Mr. Olson formerly served as district sales manager at Chicago. In his new position he will be assisted by J. Arthur Townsend and Leroy C. Rathsam, district sales engineers.

♦

Previously with General Electric Co., Frederick H. Neuhardt has joined Eastern Metal Products Co. of Tuckahoe, N. Y., and will serve as sales manager of the Industrial Division.

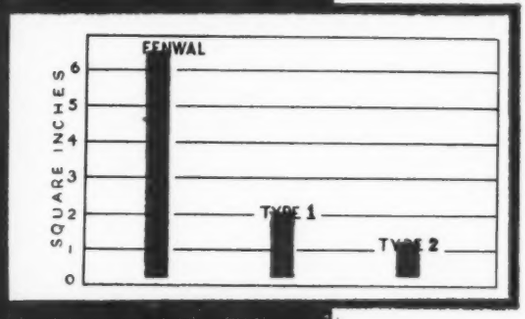
♦

Four new district sales offices have been established by Warren Telechron Co. of Ashland, Mass. These are: The Boston office, 834 Chamber of Commerce Bldg., 80 Federal St., under the direction of A. W. Pingree, district manager; the Philadelphia office, 1004 Architects Bldg., 121 South Seventeenth St., under R. J. Buckley, district manager; the Chicago office, 1440

LARGE HEAT SENSITIVE AREA SMALL HEAT STORAGE ✻

How much sensitivity do you demand from your thermostat? The high degree of sensitivity of the Fenwal Thermoswitch is due to its large heat sensitive area. This insures fast heat interchange between the medium controlled and the Thermoswitch. Its small heat storage means a minimum of heat is required to raise its temperature a given amount.

Chart shows the area of the sensitive element of the Fenwal Thermoswitch compared to Type 1 and Type 2 thermostats. Note the much larger sensitive area of the Thermoswitch.



Fenwal Thermoswitches . . . are readily adaptable for all applications . . . are easily installed even where space is extremely limited . . . are inexpensive and offer a light-weight, compact, vibration-proof, highly sensitive, yet rugged regulatory unit.

The Fenwal Thermoswitch embodies many desirable features that are not found in other types of thermostats. *When deciding on a heat control unit for* your *needs, investigate Fenwal Thermoswitches. Send for the Thermotechnics booklet including "Fourteen Facts in Fenwal's Favor."*

✻ #2 of the "Fourteen Facts in Fenwal's Favor".

Fenwal

FENWAL INCORPORATED

37 PLEASANT STREET

ASHLAND MASSACHUSETTS

THERMOTECHNICS FOR COMPLETE TEMPERATURE REGULATION

LOW COST Rust Prevention!

USE HARPER NON-FERROUS AND STAINLESS FASTENINGS

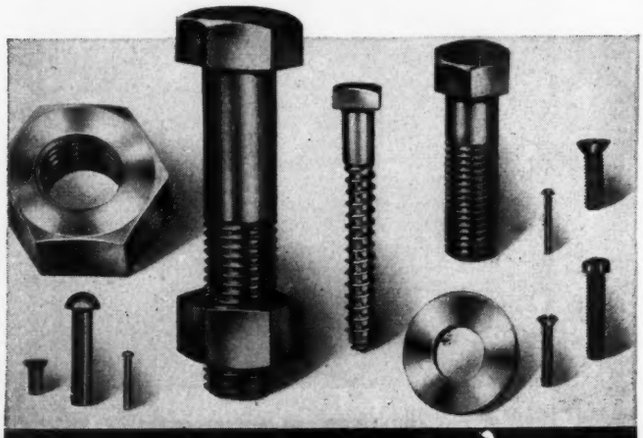
Rust is the Great Destroyer. Every year it causes damage of "war debt" proportions.

Fortunately, the cost of preventing rust and corrosion through the use of Harper Everlasting Fastenings is low. Of course, the first cost of a bronze bolt or a stainless screw is more than a comparable fastening made of common steel. Yet the difference in price is small, particularly when considered in relation to the total cost of a machine, instrument or other fastened assembly. Everlasting fastenings add longer service life to your product...and the ability to perform under tough conditions. Such qualities provide a big advantage over competition.

4360 ITEMS IN STOCK

Harper is known as "Headquarters for Non-Ferrous and Stainless Fastenings" . . . carries large and complete stocks of 4360 different items and is continually adding others . . . maintains large stocks of metals in bars, rods, wire, sheet and other basic forms from which special fastenings can be quickly made. Write for 1945 Catalog.

THE H. M. HARPER COMPANY
2626 Fletcher Street • Chicago 18, Illinois
BRANCH OFFICES:
New York City • Philadelphia • Los Angeles • Milwaukee • Cincinnati • Houston
Representatives in Principal Cities



HARPER
EVERLASTING FASTENINGS

BRASS • BRONZES • COPPER
MONEL • STAINLESS

Merchandise Mart, under J. W. Babb, district manager; the St. Louis office, 713 Ambassador Bldg., 411 North Seventh St., under H. D. Stanton, district manager.

♦

Announced by Perfex Corp., Milwaukee, are the appointments of L. B. Miller and Allen Butler as vice presidents in the Controls Division, and of I. G. Bohrman as vice president in charge of the Radiator Division.

♦

To serve as sole distributor of its standard-density Vibracork, Armstrong Cork Co. has appointed The Korfund Co. Inc. of Long Island City, N. Y., manufacturer of vibration control units of cork, rubber and steel construction. Vibracork will be furnished in two to four-inch thicknesses, heavy density being available on special order.

♦

Edward L. Taylor, general sales manager, has been elected to partnership in the Camfield Mfg. Co. of Grand Haven, Mich.

♦

With headquarters in Atlanta, Ga., William E. Thomas has been appointed division sales manager in charge of valve sales in the Southeastern states for Homestead Valve Mfg. Co. of Coraopolis, Pa. He was previously Southeastern division manager for Chicago Metal Hose Co.

♦

Plans have been made by Pittsburgh Corning Corp. for a $300,000 expansion of existing facilities at its Port Allegany, Pa., plant for the manufacture of a glass insulation material known as Foamglas.

♦

Among other appointments recently made by Ellinwood Industries, Los Angeles, is that of H. R. Edwards as regional sales manager of the eastern region of the Farm Equipment Division. Mr. Edwards will cover the Atlantic seaboard, part of the middle West and deep South. Albert Rebel has been named export sales manager of the division.

♦

Previously with General Controls, John Hammond has joined the Allied Control Valve Co. Inc. of South Norwalk, Conn., and will serve as vice president and general manager.

♦

Eaton Mfg. Co. Cleveland, has acquired all outstanding shares of Dynamatic Corp. of Kenosha, Wis., manufacturer of Dynamatic electrical induction devices for application to drives, couplings, brakes, dynamometers, and electronic controls.

♦

Addition of Robert H. Ehret to the headquarters sales staff has been announced by The Electric Products Co., 1725 Clarkstone Road, Cleveland 12.

♦

With offices in New York, Ellsworth H. Sherwood has succeeded Charles Gaspar as New York and New England district manager for the railway sales division of National Malleable & Steel Castings Co. In addition to his new duties, Mr. Sher-

ONE MORE STRAW—ONE LESS CAMEL!

Many centuries ago the weight of one last straw broke a camel's back. Thus somebody's artless optimism about the strength of a vital part wrecked a perfectly good materials handling system.

Today it is plain, cold fact that ignoring the depressing effect of low temperatures on the impact strength of vital steel parts can be a short cut to a long line of avoidable, costly troubles.

The solution for the problem in the fable was less load or more camel. For the modern problem the answer is a molybdenum steel that combines the deep hardening and freedom from temper brittleness necessary to provide good low temperature impact strength.

MOLYBDIC OXIDE—BRIQUETTED OR CANNED • FERROMOLYBDENUM • "CALCIUM MOLYBDATE"
CLIMAX FURNISHES AUTHORITATIVE ENGINEERING DATA ON MOLYBDENUM APPLICATIONS.

Climax Molybdenum Company
500 Fifth Avenue · New York City

MOLY

1" I.D. Stainless Steel Hose, closed pitch, made from Type 304 Carpenter Stainless Tubing 1½" O.D. x .025" wall. Used for fluid and steam lines, flexible connections between rigidly held units, to reduce vibration.
Why the Fabricator Made it from Carpenter STAINLESS TUBING

● This flexible stainless steel hose is one of many specialty items made by fabricators of Carpenter Stainless Tubing. When *you* need specialty Stainless Tubing to meet your design-engineering requirements, make use of our experience. We can provide the correct type of close tolerance 100% hydrostatically tested tubing. And we will be glad to put you in touch with specialty fabricators.

ASK FOR THIS NEW SLIDE CHART OF TECHNICAL DATA

In convenient form, this slide chart gives you useful information on: • Mass Velocity Constants (V_m) • Velocity Constants (V_l) • Cross Sectional Areas of Tubes • Sq. Ft. Internal Surface per Lineal Foot of Tube • Physical Properties of Stainless Tubing. A note on your company letterhead will start your Stainless Tubing Slide Chart on its way. Write for your copy today.

Welded Alloy Tube Division
THE CARPENTER STEEL COMPANY
Kenilworth, New Jersey

Carpenter
WELDED
STAINLESS TUBING

wood will continue in an advisory capacity in wheel promotion. Floyd Snyder has been promoted to sales agent at the New York office, while George F. Wilhelmy has been advanced to sales agent at the Cleveland office.

♦

Appointment of A. C. Weiss as special constultant in plastics engineering and design has been announced by Barnes & Reinecke, designers and engineers of Chicago.

♦

Previously with Westinghouse Electric Corp. in Mansfield, O., W. C. Stevens has formed the Stevens Mfg. Co. Located in Mansfield, the new company will make bimetallic thermostats for general industrial use as well as for such applications as flat irons, ironing machines and sterilizers.

♦

Metal Hydrides Inc., Beverly, Mass., has appointed Dr. D. S. Eddelsheimer as sales manager and chief physical metallurgist. Previously Dr. Eddelsheimer was in charge of a research project at the University of New Hampshire.

♦

Change of name to The Polymer Corp. has been announced by the Reading Division of Atlantic Plastics Inc., producers of Nylon and other specialized thermoplastics in the form of rods and heavy section moldings. The new company will be an affiliate organization instead of a division.

♦

To succeed F. L. Schneider, the American Welding & Mfg. Co. has appointed W. D. Cleavenger as Chicago district manager, with headquarters at 332 South Michigan Ave. Mr. Schneider has joined the Wallace Supplies Mfg. Co.

♦

A new Special Alloy Division has been formed by Jessop Steel Co. of Washington, Pa., to advise users of alloy steels in their reconversion programs.

♦

Opening of the St. Louis Technical Section of its Development and Research Division has been announced by The International Nickel Co. Inc. Located at Room 810 of the Ambassador Bldg., 411 North Seventh St., the new section is under the direction of George A. Fisher Jr. Mr. Fisher will furnish technical assistance in the territory covering Missouri, the southern half of Illinois, the southwestern section of Iowa, the northern half of Arkansas, the wester portion of Tennessee and the states of Nebraska and Kansas.

♦

Previously connected with Bendix Aviation Corp., Edgar P. Hollister has been appointed manager of the New York factory branch of General Controls Co., Glendale, Calif. He will serve users of automatic pressure, temperature and flow controls in New York state, New Jersey and Connecticut.

♦

Establishment of a development engineering department has been announced by the Rustless Iron & Steel Division of American Rolling Mill Co. Stanley P. Watkins, previously manager of the Rustless market development department, will head the new department. He will be aided by Thomas L. Moore who has been named assistant manager. Formerly Mr. Moore was

It has tremendous capacity for its size!

Where there's a need for high unit capacity, Torrington Needle Bearings furnish the answer to a great number of design problems... one reason also why they serve in such a wide range of applications in so many diversified industries.

Behind this outstanding advantage of tremendous radial load capacity for their size lies the principle of needle bearing design: the full complement of small diameter precision rollers which provides many linear inches of bearing contact surface... assures load distribution and minimizes wear.

There are many other Needle Bearing features important to you as designer, manufacturer or user of mechanical equipment... ease of operation under severe conditions... ability to retain and distribute lubricant evenly over bearing surfaces... low initial cost... and unit construction permitting simple, rapid installation.

If you are not fully acquainted with all the advantages which make Torrington Needle Bearings applicable to *your* product, write for our informative Catalog 32... or for prompt assistance on immediate friction problems confronting you, consult our engineering staff.

•

THE TORRINGTON COMPANY

TORRINGTON, CONN. • SOUTH BEND 21, IND.

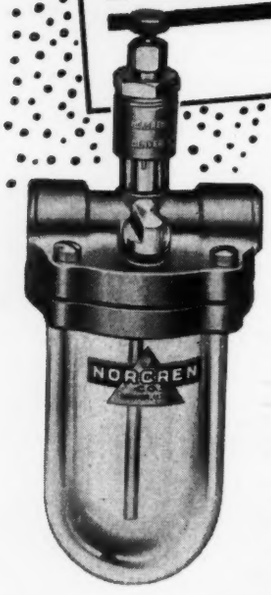
Offices in All Principal Cities

TORRINGTON NEEDLE BEARINGS

FOR AIR CYLINDERS

A LITTLE OIL . . .

A LOT OF PROTECTION!



Oil introduced into the air stream forms an oil-fog, which travels with the air, coating every surface it touches with a protective film of oil. Simplest, most positive, economical, fool-proof method known for lubricating air driven tools and cylinders.

Norgren Lubricators, operating on this principle, work when the tool works. Automatically inject exactly the right amount of clean oil into the air that drives the tool. Coat all moving parts of the tool or cylinder with a film of oil for faster, smoother performance. Protect against rust and corrosion when tool is idle. Any desired feed may be used without loss of pressure.

Retain the brilliant performance built into machine tools—with a Norgren Lubricator.

Write for Catalog No. 400. C. A. Norgren Co., 220 Santa Fe Drive, Denver 9, Colorado

Norgren

western sales manager. Also announced by Rustless Iron & Steel is the appointment of Peter B. Kline, previously eastern sales manager, as assistant manager of stainless bar and wire sales.

♦

TEC Pencil Co. has moved to its new factory located at 3512 Helms Ave., Culver City, Calif.

♦

Associated with the company since 1935, F. R. MacFadyen has been transferred from the Milwaukee office to the engineering department of The Timken Roller Bearing Co. in Canton, O. Serving as industrial engineer, Mr. McFadyen will assist S. M. Weckstein and Paul Haager.

♦

Dwight Adams, formerly with the Heppenstall Co., has been named to head the Philadelphia office of Pittsburgh Steel Foundry Corp. of Glassport, Pa. He will also represent the Fort Pitt Castings Division of the company.

♦

Previously district manager of the Washington office of Ampco Metal Inc., Milwaukee, R. S. Stover has been appointed supervisor of the Philadelphia-Washington district, with headquarters at the Wilford Bldg. in Philadelphia. Elmer E. Whitson will remain as Philadelphia district manager.

♦

To speed the conversion of aluminum battle scrap to meet immediate building needs, the Reynolds Metals Co. has acquired the government sheet mill at McCook, Ill.

♦

Change of name to P.I.B. Products Inc. has been announced by Polytechnic Research & Development Co. Inc., Brooklyn, N. Y. Also announced is the opening of a consulting engineering laboratory. The company is headed by Dr. H. S. Rogers, president of the Polytechnic Institute of Brooklyn, and is under the technical direction of F. J. Gaffney, previously in charge of measurement and test equipment development at the MIT Radiation Laboratory.

♦

Following the resignation of H. C. Stults, Wickwire Spencer Steel Co. has appointed Harry Bottomley as sales representative in the southern Pennsylvania and Virginia territory.

♦

Allied Control Co. Inc., New York, has acquired the B. F. Miller Co. of Trenton, N. J., manufacturer of all types of transformers. B. F. Miller will remain in charge as executive vice president and general manager of the company.

♦

Formerly manager of the branch office in Cincinnati, Bernie G. Silberstein has been promoted to Ohio Valley district manager of Ilg Electric Ventilating Co. He will supervise engineering and sales activities in branch offices in Cincinnati, Columbus, Louisville, Knoxville, and Charleston, W. Va.

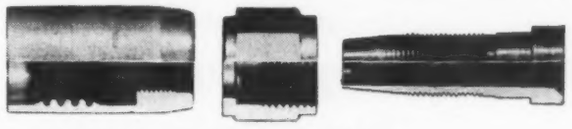
♦

Appointment of Bruce A. Irwin as sales manager has been announced by the Hammel-Dahl Co. of Providence, R. I., manufacturer of automatic control equipment.

To Industry...
USE AEROQUIP FLEXIBLE HOSELINES WITH DETACHABLE AND REUSABLE END FITTINGS,* Make Your own hose assemblies to desired lengths with Aeroquip Hose and Detachable Fittings. Your customer will like the product better when it is equipped with Aeroquip Hose Assemblies, because idle time will not be a problem . . . just a few minutes to reassemble the fittings on a new length of hose and your equipment is back on the job.
ELIMINATE DOWN TIME—

AEROQUIP FIRSTS

Self-Sealing Couplings
Aeroquip X-Washers
Hydrofuse
*Another Aeroquip First



3 PIECES (Each Replaceable)

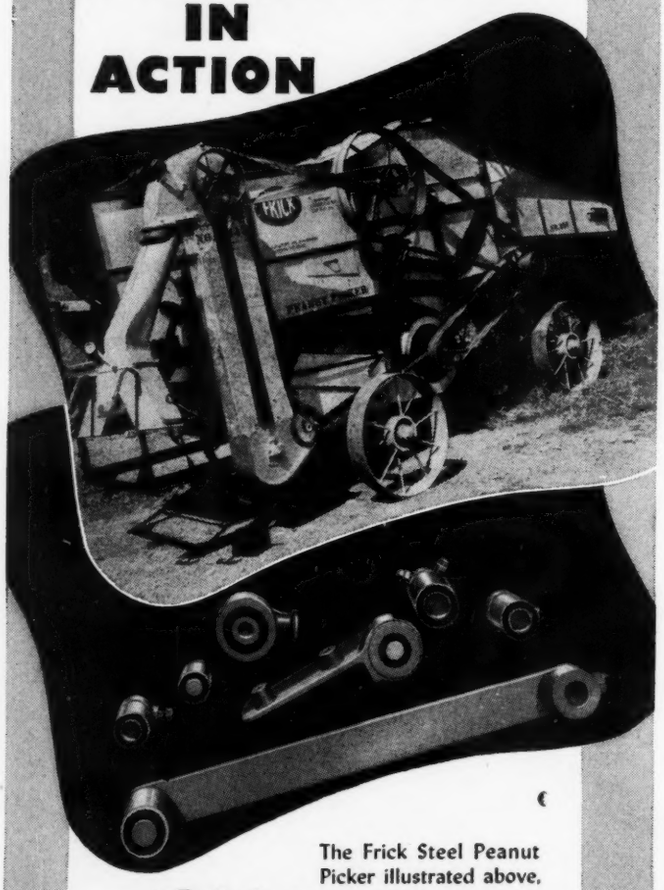
Assembly without special tools. No tightening or adjustment after assembly. Fittings can be removed from hose and reused over 100 times.

AEROQUIP CORPORATION

JACKSON, MICHIGAN

327 M & M BUILDING, HOUSTON 2 • • • 1709 W. 8TH ST., LOS ANGELES 14 • • • 803 PENCE BUILDING, MINNEAPOLIS 3

Torflex BEARINGS IN ACTION



The Frick Steel Peanut Picker illustrated above, uses 18 *Torflex* Bearings in various sizes for the ends of radius rods and pitman arms which actuate the screens.

Torflex Bearings are also used in other types of screen equipment, in rock and coal crushers and many other machines where the bearings are subject to dust conditions.

Torflex bearings require no lubrication, are practically without wear because there are no rubbing surfaces—they cushion shocks and absorb vibration. They are made in a wide variety of sizes and are applicable to all sorts of equipment.

If you have a wear or other problem in connection with any oscillating joints, we welcome the opportunity to furnish the answer.

5105 COWAN AVENUE CLEVELAND 4, OHIO

HARRIS PRODUCTS CO.

NEW MACHINES--

And the Companies Behind Them

Automotive

Mobile machine shop, Davey Compressor Co., Kent, O.
Delivery truck, Marmon-Herrington Co., Indianapolis 7.
Starter-generator type truck, U. S. Thermo Control Co., Minneapolis.
Delivery trucks, Vanette Inc., Detroit.
Long distance hauling truck, Truck Division, Eisenhauer Mfg. Co., Van Wert, O.
Rear-engined car, Bobbi Motor Car Corp., San Diego 1, Calif.
Battery charger, Motor Generator Corp. Div., The Hobart Bros. Co., Troy, O.

Construction

2-way power boom, Standard Steel Corp., Los Angeles 11.
Concrete mixer, The Jaeger Machine Co., Columbus 16.
Convertible excavator, Bucyrus-Erie, South Milwaukee, Wis.
Portable compressor, Le Roi Co., Milwaukee.
Diesel crawler-tractor, International Harvester Co., Chicago.
*Concrete mixer, Chain Belt Co., Milwaukee.

Domestic

*Radio photograph, Garod Radio Corp., Brooklyn, N. Y.
*Clothes washer, Apex Electrical Mfg. Co., Cleveland.

Industrial

Voiceless intercommunication, Simplex Time Recorder Co., Gardner, Mass.
Portable fire extinguishing unit, Cardox Corp., Chicago 1.
Marking machine, Adolph Gottscho Inc., New York 13.
Portable magnet charger, Radio Frequency Laboratories Inc., Boonton, N. J.
Motor-driven pipe cleaning machine, Spartan Tool Co., Chicago 45
*Engraving machine, Auto Engraver Co., New York.

Instruments

*Flight analyzer, Hathaway Instrument Co., Denver.

Photography

Tri-vision dual lens camera, Haneel Co., Los Angeles.
"Suitcase type" television camera and studio units, Philco Corp., Philadelphia.
*8 mm. movie projector, De Jur-Amsco Corp., Long Island City, N. Y.
*Radar camera, Faidchild Camera & Instrument Corp., Jamaica, N. Y.
*X-ray camera, Fairchild Camera & Instrument Corp., Jamaica, N. Y.

Printing

*Automatic cylinder press, Banthin Engineering Co., Bridgeport, Conn.

Rubber

Shredding machine, Jeffrey Mfg. Co., Columbus 16, O.

Testing

External comparator, Pratt & Whitney Div., Niles-Bement-Pond Co., West Hartford, Conn.
Electrodynamic vibration exciter and calibrator, The MB Mfg. Co. Inc., New Haven, Conn.
Hydraulic tube testing press, Beatty Machine & Mfg. Co., Hammond, Ind.
Interference viewer, Optron Laboratory, Dayton 6.

Welding

*Spot welding machine, Tweezer-Weld Corp., Newark, N. J.

*Illustrated on Pages 156-159.